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THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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The Heinkel Composite Cooling System.
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METHOD TO DETERMINE THE CAVITATION FACTOR BY AIR TESTS.

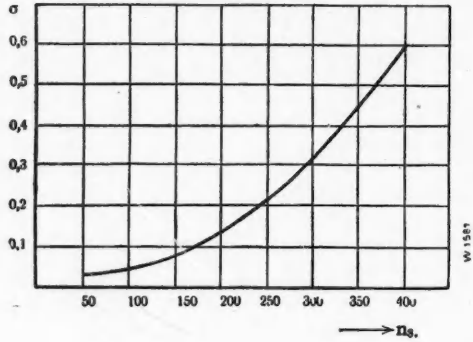
By Dr. C. KELLER and H. BLEULER. (From *Escher Wyss Nachrichten*, No. 1-2, 1939).

THE investigation of up-to-date turbines and pumps from the point of view of cavitation danger was until now effected on running models of pumps or turbines in special cavitation testing plants under pressure conditions, corresponding to the actual working conditions of the full scale machine, and by a summary determination of the admissible cavitation factor σ as defined by Thoma. By stroboscopic observation of the runner and by measuring the output and efficiency of the model turbine, the commencement of the evaporation of water in consequence of a great local drop in pressure on the surface of the runner blades could be in each case accurately observed, as well as all subsequent developments.

It is possible to distinguish several stages in the phenomenon of cavitation, and it is a matter of experience and judgment to estimate to what extent the "cavitation limit" may be approached in actual practice. As is well known, there exists a definite relation between the admissible value of σ and the specific speed n_s (compare Fig. 1).

Ackeret pointed out long ago that the values of σ obtained by the usual summary method could, in future, in view of the progress already made, only be used for preliminary and approximate estimation, and that recourse would have to be taken in a more accurate method of calculation, i.e. one that takes into closer account the actual pressures prevailing at each point of the turbine.

Fig. 1. σ_{k_r} of the cavitation limits for actual practice in dependence on the specific speed n_s.



At that time there were, however, no data available dealing with the exact distribution of pressure on the revolving blades of pumps or turbines. Solely for high-speed axial-flow wheels was it possible to obtain some general information with regard to the actual distribution of the pressures on the blades by relying upon a few measurements, carried out at Göttingen on stationary airfoil profiles and grids in connection with aviation problems. Since the distribution of pressure varies considerably with the blade pitch, the shape of the blade, and also with the degree of admission, the information was entirely insufficient for quantitative calculations.

A new method of investigation has been now developed for the first time in the laboratories of Escher Wyss. By applying this new method it is possible to determine the accurate distribution of the static pressure on the blades of a rotating runner by means of tests carried out on a model driven by air and running under the same working conditions as the true full-sized machine.

This is possible owing to the fact that, as will be shown later on, the actual distribution of pressure on the blades of the true full scale pump or turbine, is exactly similar to that of the model driven by air. Hence, we now have the means to predetermine accurately in each separate case the absolute pressures that may be expected to prevail in the true full-scale turbine. If at any point the pressure, as determined from the air test, happens to lie under the vapour pressure of water, then cavitation is sure to take place in the hydraulic turbine. If, on the other hand, the pressure is above the vapour pressure of water, there can be no danger of cavitation.

The great advantage of this method of investigation, as compared with the purely hydraulic method, lies in the fact that the places where cavitation phenomena might be expected, more particularly at the runner blades, can be much more accurately ascertained than is possible by stroboscopic observations alone. Having once ascertained the accurate distribution of pressure on the blades, the pro-

Diagrammatic graph of the pressure distribution in a low-head turbine. The model air turbine (Fig. 2) and the full scale machine (Fig. 2a) show, if the differences in level in the air turbine are neglected, exactly similar pressure distribution.

a Curve showing the distribution of pressure.
b Atmospheric pressure.
c Pressure distribution as a result of the differences in level.

0 Inlet into runner. 1 Inlet into turbine.
2 Runner outlet. 3 Suction tube outlet.
2* Point on the runner blade at which lowest pressure occurs.

Fig. 2.

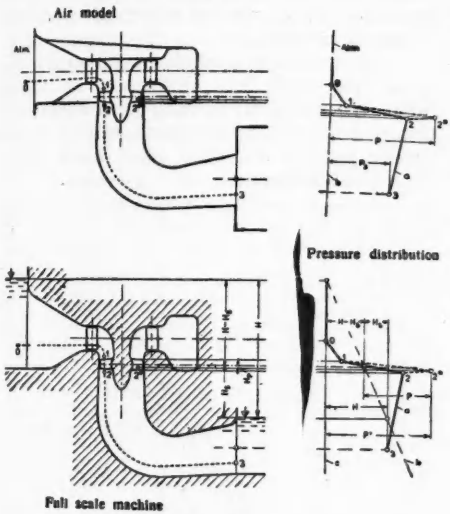


Fig. 2a.

file of the latter and the distribution of velocities can be altered in such a manner as to avoid too low pressures at any point, whereby the danger of cavitation can be reduced to a minimum.

From the aforesaid it will be evident that the new aero-dynamic method of investigation leads also in this field to new conceptions, and that it is of great advantage and usefulness to the engineer who has to deal with the investigation and construction of turbine machinery.

Pressure distribution in the model of a low-head turbine tested in an aero-dynamic plant is shown in Fig. 2, where the

1. Drop of pressure in the guide wheel is:
$$\Delta p\, 0 \div 1 = p_3 (Kc^2_m + Kc^2_{u1})$$
2. Drop of pressure in the runner wheel is:
$$\Delta p\, 1 \div 2 = 2\, p_3\, Ku \cdot Kc_{u1}$$

LIST OF SYMBOLS:

	Serie-turbine	Air-turbine model		Full scale turbine	
		Sym.	Dimen.	Sym.	Dimen.
1	Net head	P_3	mm. water	H	m. water
2	Stat. pressure on runner blade at the inlet side	P	gauge	P	gauge
3	Difference in pressure	ΔP	gauge	ΔP	gauge
4	Barometric pressure ..	b	mm. Hg.	B	gauge
5	Suction head (positive when runner is above tail race level)	h_s	m. air gauge	H_s	gauge
6	Difference in level between tail race level and suction tube outlet ..	h_3	m. air gauge	H_3	gauge
7	Difference in level between turbine inlet and tail race level	h_1	m. air gauge	H_1	gauge
8	Distance in axial direction for vertical turbines between the point of lowest pressure and the axis of runner blade .. (positive when the axis is located above that point).	$h_{p\,min.}$	m.	$H_{p\,min.}$	gauge

3. Head regained in suction tube, ηs being the efficiency of the suction tube, is:
$$\Delta p\, 2 \div 3 = \eta s \cdot Kc_2^2 \cdot p_3$$

The differences in level h may be neglected when using air as the testing medium, and they can be entirely eliminated when the axis of the turbine is arranged horizontally. The head H of the actual turbine is replaced by the pressure p_3 in the air tests, this pressure corresponding to the difference in pressure between the surrounding atmosphere and the equalizing chamber following the suction tube. It may be mentioned that this pressure p_3 is only some hundreds mm. water gauge in magnitude.

The relation between the lowest measured pressure $P_{min.}$ and the measured difference in pressure p_3 should be defined as:

4. $$\delta = \frac{P_{min.}}{P_3}$$

Since the velocities in the air test are kept within such limits that the compressi-

bility of the air can be neglected, the laws for series-turbines may be applied to the air turbine as also to the full-size turbine, and consequently the heads of the full-size turbine can be determined in the following manner.

In the above equations 1 to 4, the total head H can be substituted by p_s. The differences in pressure thus obtained are either increased or reduced by the amount of positive or negative differences in level present in the corresponding part of the turbine. The particular distribution of pressure for the corresponding model of the full-size turbine as shown in Fig. 2 is then obtained, the resulting differences in pressure head being:

5. Head race level at the inlet $H - H_1$.
6. Drop of pressure in guide wheel:
$$\Delta P_{0 \div 1} = H(Kc^2_m + Kc^2_{u1}) - H_1 + H_s$$
7. Drop of pressure in the runner wheel:
$$\Delta P_{1 \div 2} = 2H \cdot K_u \cdot Kc_{u1}$$
8. Pressure regained in the suction tube:
$$\Delta P_{2 \div 3} = \eta s \cdot Kc^2_2 \cdot H + H_s - H_3$$

Since the law for the series-turbine is applicable, the pressure distribution should be exactly similar to that of the aerodynamic model. Designate further the lowest pressure present in this assumed pressure distribution with $P'_{min.}$ Then the factor δ as defined above is a constant for the series-turbine

9. $$\delta = \frac{P'_{min.}}{H}$$

In order to obtain the real static pressures of the full-size turbine, the level differences which decrease in a linear manner with the height above tail race level should be added to the assumed pressure distribution, as may be clearly seen from Fig. 2a.

The actual lowest pressure is then given by:

10. $P_{min.} = H - H_s - P'_{min.}$

if the small difference in level $Hp_{min.}$ between the point of lowest pressure and the axis of the runner blade is at first neglected.

Cavitation sets in when $P_{min.} = P_{vapour} - B$, or approximately when:

11. $P_{min.} = -B$

By introducing this equation in 10 and rearranging the latter equation, it is:

12. $$\frac{P'_{min.}}{H} - 1 = \frac{B - H_s^*}{H}$$

where H_s^* is the suction head at which cavitation sets in. In the right side of equation 12 appears the well known Thoma's cavitation factor:

13. $$\sigma^* = \frac{B - H_s^*}{H}$$

Hence, with equation 9, the simple relation is obtained:

14. $$\delta - 1 = \sigma^*$$

Thus, by substracting from the factor δ, which can be directly determined by measurement of pressure carried out in the model air turbine, the cavitation factor of the full-sized machine for the commencement of cavitation σ^* may be obtained.

For extreme low-head plants of high specific speed, the small level differences occurring within the runner wheel itself must eventually also be taken into consideration, as well as the vapour pressure P_D when using water above the normal temperature.

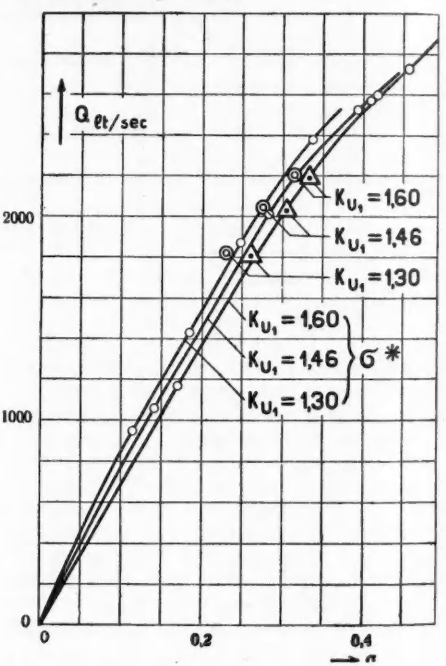


Fig. 3. The *good agreement* of the cavitation factors determined with air and with water proves the reliability of the aerodynamic method. Small circles: air test. Small double circles and triangles: Commencement of the falling off of efficiency and setting in of cavitation, as ascertained by hydraulic measurements.

The equation 14 has to be replaced by:

$$15. \qquad \delta - 1 + \frac{P_D - Hp_{min.}}{H}\,\sigma^*$$

If m is the scale of the model, then it is with sufficiently close approximation for practical use:

$$16. \qquad Hp_{min.} = m \cdot hp_{min.}$$

The exact theoretical value of $Hp_{min.}$ may be determined graphically from the pressure distribution at the surface of the runner blade.

The results of pressure measurements carried out in the manner described above in an aero-dynamic testing plant on a model of Kaplan turbine are shown in Fig. 3 in the usual way: Maximum flow of water Q in dependence on σ^*. Into this Fig. 3 are also plotted some values of σ^* that have been obtained on the hydraulic cavitation testing plant in a turbine similar to the model, and also σ critical values, i.e., values at which the efficiency falls off spontaneously. It will be seen that the values obtained by the two methods show relatively good agreement. Detailed investigations of this kind are of special importance with regard to the development of Kaplan turbines for high heads, since in this case the back pressure must be kept as low as possible, in order to reduce the work of excavations.

RING SPRUNG UNDER-CARRIAGE LEGS.

By DR. ING. E. RICHTER. (From *Deutsche Motor-Zeitschrift*, No. 6, Vol. XVI).

THE **Kronprinz** under-carriage legs are sprung with a system of ring springs as shown in Fig. 1. The double tapered outer surfaces of the inner rings rest against the double conical inner surfaces of the outer rings. Thus a force acting on the spring compresses the inner and elongates the outer rings, whereas a possible shock will be to a large extent absorbed by the friction between the tapered surfaces. This spring may be compressed only until the spring rings become to lie on each other and thus any over stress can be easily avoided. At the same time the rings are subjected only to compressive, respectively to tensile stress, enabling to use the material to the fullest extent under the most favourable stress conditions. With a view to the latter fact the weight of a **Kronprinz** spring is claimed to be 3 to 4 times lighter, as compared with a helical spring of equal strength.

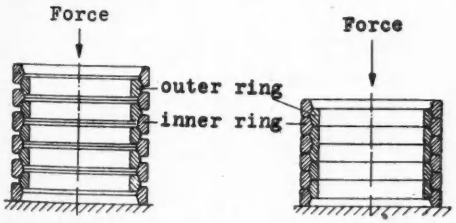


Fig. 1.

Fig. 2 shows the operation of the **Kronprinz** under-carriage legs, sprung with the above described springs. The landing shock is transmitted to knobs **2** and **3** and, after absorbing a part of the landing shock, the spring softly recoils, pushing down knob **3**. The force is transmitted from knob **3** through tubes **4** and **6** and rim **8** to spring **1**, which is being held at the other end by anchor **7**.

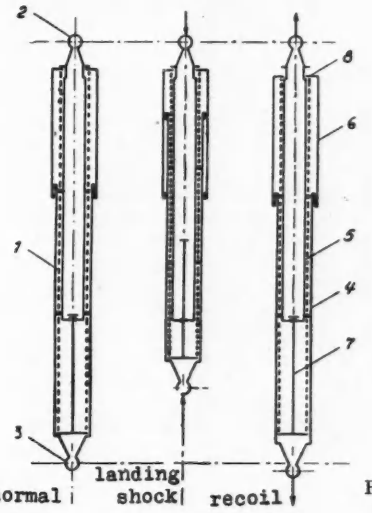


Fig. 2

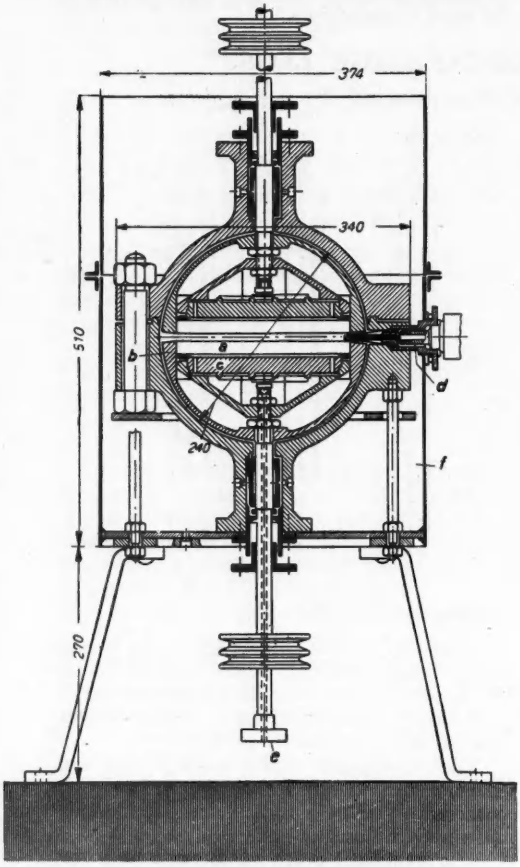
The **Kronprinz** under carriage legs are claimed to need very little maintenance, to be insensitive against possible damage, and to be unaffected by heat or cold.

The aeroplane **Ju 52**, which carried out the Himalaya expedition in 1937 under difficult conditions, is said to have been equipped with the above under-carriage legs, which proved to be very satisfactory.

THE IGNITION LAG IN DIESEL ENGINES.

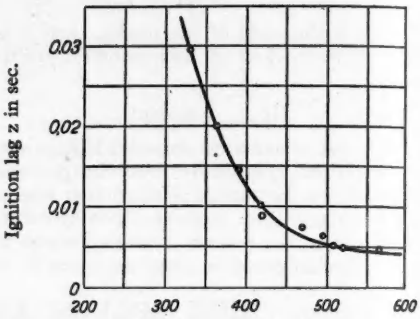
By DIPL. ING., PH. D., H. H. WOLFER. (From *Sulzer Mitteilungen*, No. 3, 1939).

THE experiments to study the ignition lag were carried out in a vessel, as shown in Fig. 1, in which the influence of air turbulence, too, could be determined. In a spherical cast steel vessel two shells with inserted heating plates form the combustion chamber, corresponding to that of an opposed piston engine. The shells are fixed on two shafts driven by belts from an electric motor. The rotation of the shells is thus soon transmitted to the air, the speed and direction of the air swirl being at the same time also determined. The heating wires are introduced through the shafts and an oil bath is provided to cool the vessel.



a combustion chamber ; *b* rotating shell ; *c* heating plates ; *d* injection valve ; *e* contact for heating wire ; *f* oil bath.

Fig. 1. Experimental combustion vessel. Dimension in mm.

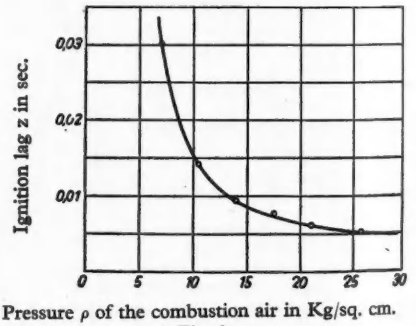


Temperature σ of the combustion air in deg. C.

Fig. 2.
Ignition lag as a function of the air temperature.

Flat steel tubes rolled in a spiral line were used as indicators for the gas and fuel pressure and the results were recorded on a card pasted on an electrically driven drum. The gas temperature was gauged with a thermo-element. In connection with this it was also pointed out that as the vessel's walls were mostly red hot during the experiments, the bright screen surface became oxydized. On the other hand because this could not be prevented, the temperatures recorded were somewhat too high.

The experiments carried out showed that the ignition lag z was mostly influenced only by the temperature and pressure during the combustion, as shown in Figs. 2 and 3. No



Pressure ρ of the combustion air in Kg/sq. cm.

Fig. 3.
Ignition lag as a function of the air pressure.

other factors seemed to have any marked effect, thus pointing to the correctness of the chain reaction theory.

According to the chain reaction theory, as developed by N. Semenoff and others, the reaction speed is proportional with the number of the activated molecules and is influenced by:

The creation of activated molecules through collision of slower molecules.

The disappearance of activated molecules through scattering their energy owing to collisions.

The disappearance of activated molecules owing to their entering into reaction.

The appearance of new activated molecules in consequence of the reaction.

It is stated to follow from this that with increasing thermal effect due to the reaction and with increasing inclination of the molecules to become unstable, the reaction chain lengthens and the reaction is accelerated. This is further supported by increasing gas pressure, as the latter is inversely proportional to the mean free paths of the molecules, causing more frequent collisions. The probability of the collisions appears also to be influenced by the concentration of fuel, oxygen and inert gases. Finally, the development of the reaction chain seems to depend, too, upon the form of the combustion space.

Thus the speed of the reaction may be expressed as:

$$w=A\,(e^{\sigma t}-1),\ \text{or}\ A\,(1-e^{\sigma t})$$

where t is the time which passed since the beginning of the reaction, A and σ depending upon the factors mentioned before.

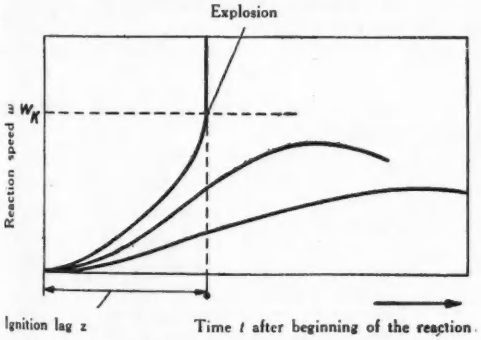


Fig. 4. Speed of reaction for reactions taking place with and without explosion.

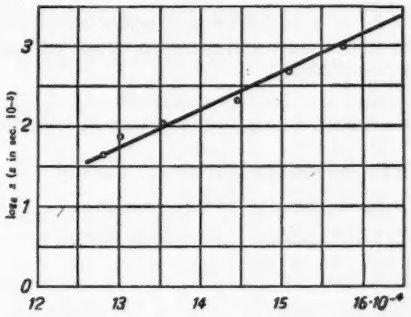


Fig. 5. Logarithm of the ignition lag, $\log z$, as a function of $10^4/T$.

As the reaction accelerates, the equation shortly becomes:

$$w=A.\ e^{\sigma t},$$

indicating that an explosion may occur without any previous rise of temperature. Further, according to Nernst, under certain conditions the heat developed by the reaction might not be completely abstracted and then it raises further the combustion temperature. This in its turn increases the reaction speed and may also lead to an explosion. Thus an explosion seems to be influenced simultaneously by the chain reaction as well as by thermal conditions.

Assume now that the heat removed from the combustion chamber is constant. In this case the reaction speed w increases first according to the laws mentioned before; but if a certain critical speed w_k is reached, the heat developed is equal, or more than what can be abducted, and a thermal explosion will happen, as shown in Fig. 4. When this critical speed is not actually reached, the reaction speed slackens gradually owing to the decreased concentration of reacting molecules.

This critical reaction speed w_k has been found to be constant for a particular combustion chamber, and the time required to reach this critical speed is the ignition lag z. Thus it is for the critical speed:

1. $$w_k=A.\ e^{\sigma z}$$

and according to Semenoff for long chains

2. $\sigma=\text{const.}\ p^n.\ e^{-E/RT}$, where

p=pressure
E=energy possessed by the average molecule
R=Gas constant
T=Absolute temperature
n=constant

As A is stated to vary rather slowly with the temperature and pressure, equation(1) may be regarded as :

3. $w_k = \text{const.}\ e^{\sigma z}$

or because w_k is also constant in logarithm

4. $\sigma z = \text{const.}$

and substituting equation (2) into (4)

5. $p^n \cdot e^{-E/RT} \cdot z = \text{const.}$

Thus the ignition lag z will be from (5)

6. $z = \frac{c_1}{p^n} e^{c_2/T}$

where the constant in (4) is designated by c_1 and E/R by c_2.

From equation (6) it is for constant pressure

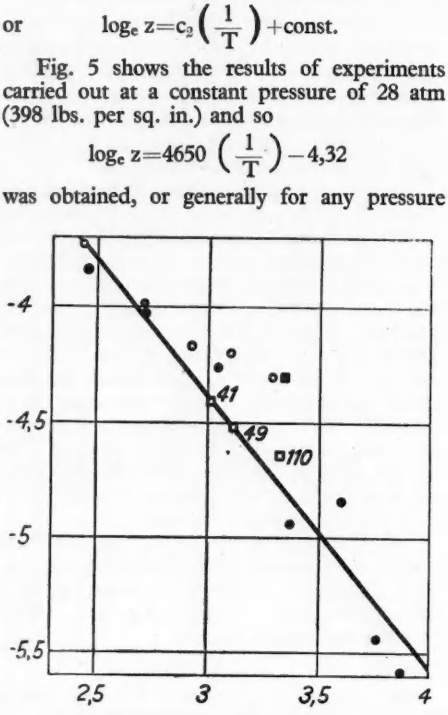
$$z = \text{const.}\ e^{c_2/T}$$

or $$\log_e z = c_2 \left(\frac{1}{T}\right) + \text{const.}$$

Fig. 5 shows the results of experiments carried out at a constant pressure of 28 atm (398 lbs. per sq. in.) and so

$$\log_e z = 4650 \left(\frac{1}{T}\right) - 4{,}32$$

was obtained, or generally for any pressure



$\log_e \rho$ Cp in kg./sq. cm.

■ Tests 1 to 7. ● Tests 14 to 20.
○ Tests 8 to 13. □ Other tests.

Fig. 6.

Logarithm of the constant c, $\log_e c$, as a function of the logarithm of the air pressure, $\log_e p$.

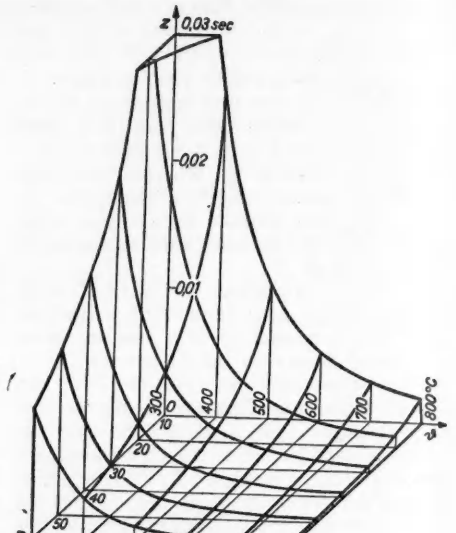


Fig. 7.

Ignition lag z as a function of the air pressure and of the air temperature.

7. $z = c \cdot e^{4650/T}$

where with reference to equation (6) c varies according to

8. $c = c_1/p^n$

or in logarithm

$$\log_e c = \log_e c_1 - n \cdot \log_e p$$

Fig. 6 shows the results of the experiments carried out and as a result

$$\log_e c = -0.83 - 1.19 \log_e P$$

was obtained from this

9. $$c = \frac{0.44}{p^{1.19}}$$

Thus the final expression for the ignition lag was found to be

10. $z = \frac{0.44}{p^{1.19}} \cdot e^{4650/T}$, where

z=ignition lag in sec. 10^{-3}

p=combustion pressure in kg/sq. cm.

T=absolute combustion temperature in deg. C.

The above equation is illustrated in Fig. 7.

Tests on an engine are claimed to have confirmed the correctness of equation (10) for all fuels with a cetene number greater than 50.

DETAILS OF DESIGN No. 5 BLAST FURNACE INLAND STEEL Co.

By H. W. JOHNSON (From *Iron and Steel Engineer*, April, 1940).

FOUR areas of construction are discussed. First is the equipment provided to prepare the coke and charge it into the furnace. Next are the factors in design which determine the radial distribution of the gases in the stock column. The third is the height of the furnace, and the fourth the provisions made for heating and controlling the blast.

Coke Handling.

In Lake Superior practice, coke is the material largest in size, and occupies 55 to 65 per cent. of the volume of materials charged into the furnace. It is obvious, therefore, that since it occupies most of the volume and is the largest in size, it provides a large proportion of the voids in the furnace which are necessary for the gas flow. Unfortunately, coke is a brittle product, and the size is easily reduced by handling. The problem, therefore, is to see that the coke reaches the stock column in the best possible size.

Coke Breakage.

The fact that coke is brittle is so generally accepted that tests are commonly made to measure the amount of breakage. These shatter tests were taken on samples of coke made from 30 per cent. Pocahontas and 70 per cent. high volatile Eastern Kentucky coal, coked in a 16-in. oven for 17 hours. The results shown are averages of the number of tests indicated :

Results of typical Shatter Tests showing Amount of Breakage:

Original size of sample	+4″	−4″+3″	3″+2″
Number of samples	4	4	5
Size after shatter test :			
+4 in.	21.3	—	—
+3in.	23.0	39.0	—
+2 in.	33.0	35.7	61.7
+1½ in.	15.8	18.3	30.2
−1½ in.	7.0	6.8	7.9
Original average size	4.5″	3.5″	2.05″
Average size after test	2.89″	2.60″	2.08″
Per Cent reduction	35.9	25.7	16.8

In general, coke decreases in value with decrease in size, and so every effort should be made to avoid breakage and the production of fines.

The Effect of Fine Coke on Furnace Operation.

At one blast furnace plant coke pieces less than 1¼ in. were screened out between the hoppers and the skips. It was found that the output of the furnace was greater, whilst the gross amount of coke used per ton of pig did not increase. The records (E. C. Evans, Fuel, 1927, 6-245) are given in the following table :

The Effect of Removing 1¼ in. Coke in Blast Furnace Practice.

Period	Coke arriving at works lb. per ton of pig	Coke charged lb. per ton of pig	Breeze through 1 1/4 in. screened out, lb.	Total burden	Aver. make, tons per week	Max. make, tons per week
1921	3180	3180	Nil	7560	1150	1202
1925	3120	2930	190	7570	1380	1465

Irregularity in size of materials charged into the furnace contributes to irregularity of furnace operation. A convenient method of measuring furnace irregularity is to note the difference between succeeding casts of both silicon and sulphur and determine an average. For instance in the silicon, one cast may be 1.00 per cent. and the next 1.09 per cent., with a difference of 0.09 per cent.

Variation in Iron Analyses during Test Periods:

	Sept.	Oct. 1-15	Oct. 22.31	Nov. 1-14
	%	%	%	%
Variation, silicon	.088	.097	.155	.085
Variation, sulphur	.018	.002	.0026	.025

It has been generally found that installation of coke sizing equipment has improved the uniformity in analysis of the iron produced.

Bearing in mind that the coke breaks readily, and that the resulting fines are detrimental to furnace operation, we made a particular effort in the design of this furnace to insure that the coke reached the stock column in the best possible condition. This meant that the amount of drop to which the coke was subjected must be reduced to a

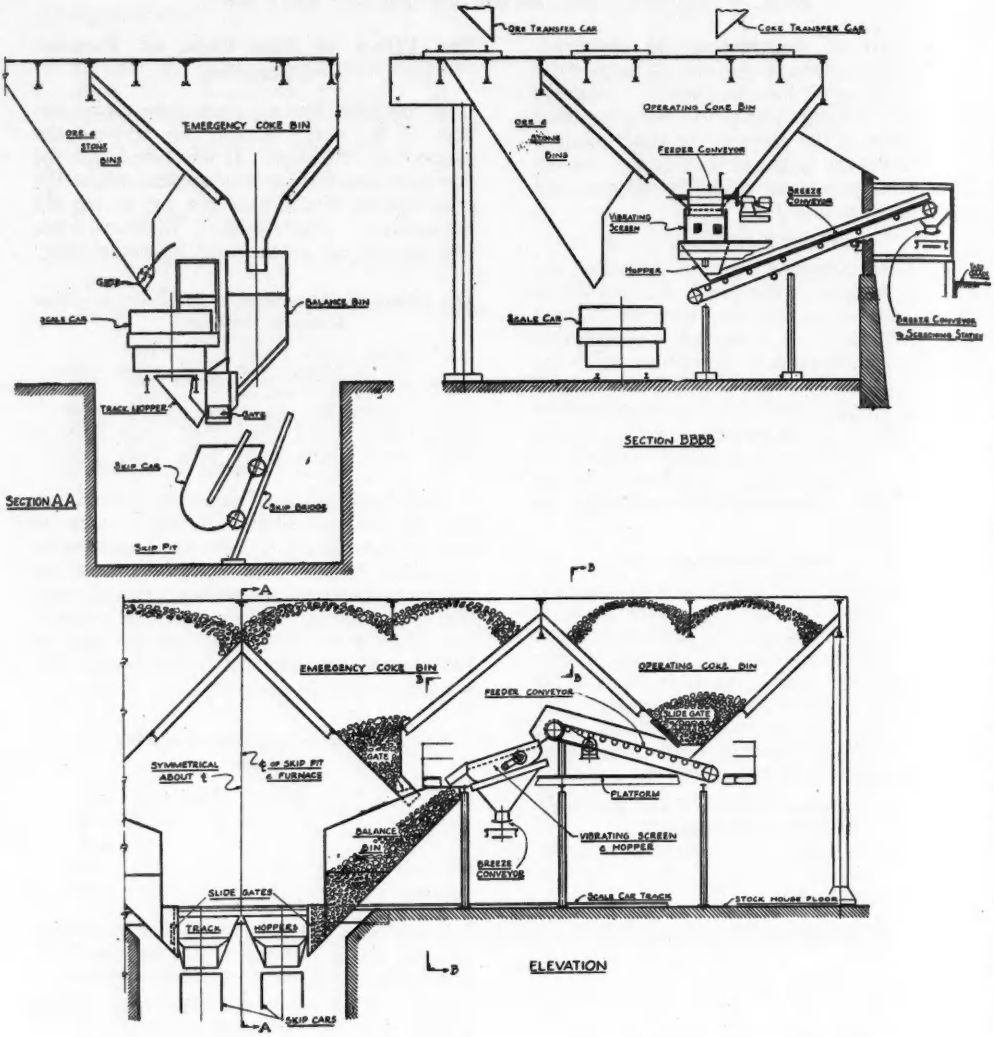


Fig. 1.

General arrangement of part of stockhouse of No. 5 blast furnace, showing the coke handling system which was designed with special consideration toward the maintenance of a uniformly sized coke.

minimum, and, second, that the coke be screened thoroughly just before charging into the skips. Fig. 1 gives the general arrangement around the skip pit showing the coke-handling system.

Factors in the Radial Distribution of the Gases and Gas Surveys.

Furnace managers are experienced in varying operation according to the materials provided, in order to achieve the best possible radial distribution.

Because we do not know what radial distribution is desired, we can only judge the desirability of a certain distribution from the efficiency of the furnace at the time. The radial surveys that have been taken on No. 5 furnace are given in the following table. From these the efficiency of the furnace can be judged, and then conclusions can be drawn as to the desirability of the measured radial distribution.

Fig. 2 shows the lines of No. 5 furnace, including most of the details of design which have an influence on the gas flow along the radius. In addition to the design there are certain factors in operation which have influence. The furnace is charged with a five skip fill—ore, ore, stone, coke, coke. The two skips of coke weigh about 10,000 lbs. The furnace is filled to within 7 ft. of the closed bell. The large bell opens in 9 seconds. The wind blown varied from 57,000 to 60,000 cubic ft. per minute.

The burden consisted of 82 per cent Lake Superior ores and 18 per cent sinter. No scrap or scale was used. All coke was screened over a 1½ in. screen as it was discharged into the skip.

No. 5 Furnace Gas Surveys

Date	Per cent CO_2																				
	W	6	12	18	24	30	36	42	48	54	60	66	72	78	84	90	96	102	108	114	120
5- 2-39	8.8	15.0	17.2	18.4	19.1	19.4	19.6	19.8	19.7	19.5	19.1	19.3	18.8	18.7	18.9	18.8	18.3	...	16.5	15.8	13.6
4	15.0	13.1	14.8	16.1	17.0	18.3	18.2	17.3	17.6	19.2	19.5	19.4	19.2	19.3	19.5	19.3	18.6	17.9	16.0	12.3	11.9
5	7.8	6.9	13.2	14.7	17.1	17.0	17.3	17.6	18.3	18.4	18.7	19.2	19.1	19.3	17.6	15.8	14.6	14.2	13.1	10.2	9.4
9	12.0	11.3	11.9	14.1	17.4	18.1	16.8	18.1	18.0	18.3	18.4	18.2	16.1	15.1	15.0	13.6	13.3	11.3	10.3	9.1	12.0
23	8.9	13.3	15.8	17.0	17.1	18.0	17.9	17.8	18.2	18.3	18.2	18.1	19.4	17.4	17.3	17.0	16.8	15.4	12.4	11.7	11.6
6- 5-39	12.7	12.8	12.3	13.1	15.9	17.1	17.4	17.9	17.9	17.9	18.1	18.2	18.3	18.3	17.9	17.0	16.4	16.5	11.0	9.9	9.9
6	10.0	12.3	13.3	14.4	16.6	17.8	18.3	18.7	18.5	18.8	18.7	18.6	18.0	17.7	16.7	16.0	14.4	12.5	10.8	9.0	9.9
7	11.0	13.7	16.3	19.0	19.0	19.2	19.1	19.0	19.0	19.1	19.0	18.3	17.5	15.8	16.0	15.2	13.1	...	11.4	12.0	8.7
8	9.0	11.1	13.0	15.5	16.6	18.5	18.5	18.4	18.4	18.4	18.6	18.2	...	17.4	17.4	16.8	15.3	13.3	11.8	10.0	9.2
9	10.2	10.9	14.0	19.1	19.3	19.7	19.0	19.3	19.3	18.6	18.2	18.1	16.6	14.6	13.3	12.1	10.3	5.1	3.7	2.9	4.6
19	5.7	9.6	10.0	15.3	17.3	17.9	18.7	19.4	20.0	20.3	19.7	19.4	19.2	17.7	17.7	16.7	13.3	11.1	8.0	6.9	7.6
19	4.5	6.6	8.7	10.5	16.2	15.9	16.6	17.5	18.4	19.2	19.5	19.8	18.3	18.6	16.6	14.8	...	14.0	10.5	9.5	11.7
Average	9.6	11.4	13.4	15.6	17.4	18.1	18.1	18.4	18.6	18.8	18.8	18.7	18.2	17.5	17.0	16.1	14.9	13.1	11.3	9.9	10.0

	Per cent CO																				
5- 4-39	28.2	28.8	27.2	25.3	24.2	21.6	21.6	21.8	21.3	20.1	19.6	20.0	20.0	20.1	19.2	19.8	19.6	19.7	21.0	25.4	26.0
5	31.0	...	25.4	25.8	22.3	22.5	22.4	22.1	22.2	22.1	21.3	20.3	20.7	21.7	22.9	25.9	27.8	28.6	30.5	34.4	34.6
9	26.0	25.9	26.1	23.8	21.7	21.1	19.8	20.2	20.6	20.5	20.6	20.3	23.5	24.9	25.4	27.4	27.9	29.1	31.3	31.9	31.4
23	27.3	23.7	21.2	21.0	20.0	19.4	19.3	19.4	18.9	18.7	19.0	19.1	19.0	19.6	19.3	20.2	20.8	22.6	26.6	28.3	28.4
6- 5-39	30.6	30.0	30.5	28.9	24.7	22.8	22.4	21.7	21.6	22.3	21.7	21.7	21.6	21.5	22.1	22.7	23.1	23.5	28.6	29.7	29.7
6	28.9	26.9	25.9	24.6	23.2	22.1	21.9	22.3	22.0	21.5	21.3	21.6	21.8	22.1	23.1	24.0	25.8	27.9	30.0	32.3	31.1
17	30.0	26.5	23.5	21.8	21.8	21.8	21.7	21.8	20.6	20.7	21.8	22.7	24.1	27.8	26.6	27.8	30.7	...	32.6	32.0	34.7
18	31.2	28.1	26.0	24.0	23.4	22.6	22.7	22.6	22.3	22.6	22.7	23.2	...	24.7	25.3	25.3	27.6	29.6	31.7	33.5	33.7
19	30.8	27.6	26.2	21.1	20.9	21.3	21.2	21.3	21.3	21.8	22.2	22.3	23.8	25.8	27.7	28.9	30.9	36.9	38.3	38.5	37.4
19	35.9	30.6	28.8	24.6	22.8	22.3	21.3	20.8	20.0	19.6	20.2	20.6	20.6	22.3	22.3	23.6	27.7	29.7	33.7	35.7	34.8
19	36.2	33.6	31.3	29.0	23.3	24.1	23.6	22.9	22.0	20.5	20.5	20.3	19.7	21.4	23.4	25.2	...	26.5	30.9	32.3	27.7
Average	30.6	28.2	26.6	24.5	22.6	22.0	21.6	21.5	21.2	20.9	21.0	21.1	21.5	22.9	23.4	24.6	26.2	27.4	30.5	32.2	31.8

Height of Furnace.

The proper height of a furnace is still a matter of discussion, although the recent trend has been definitely towards greater height. Barrett (Design and Operation of a Modern Blast Furnace, *Iron and Steel Engineer*, February, 1939) discussed this problem and recommended that the modern furnace should not be less than 100 ft. From the empirical aspect, in general, the greater the height of the furnace, the greater the amount of hanging and slipping, other things being equal. The irregular movement of the stock is very damaging, so what is gained by added height is frequently lost by the greater furnace irregularity. But as the diameter of the furnace is increased, there is less friction on the walls per unit of volume, less chance of arching, and it is possible to build the larger furnace higher without experiencing any more irregularity.

Heating of the Blast.

Our experience has shown that stoves of standard basket weave construction with a $2\frac{1}{2}$ in. brick thickness and a $5\frac{1}{4}$ in. checker opening, withstand heat successfully with fillers in these flues up to 6 ft. from the top. Fig. 3 is a photograph of a stove of this design, which was built in February, 1936, and had been in actual service for 31 months.

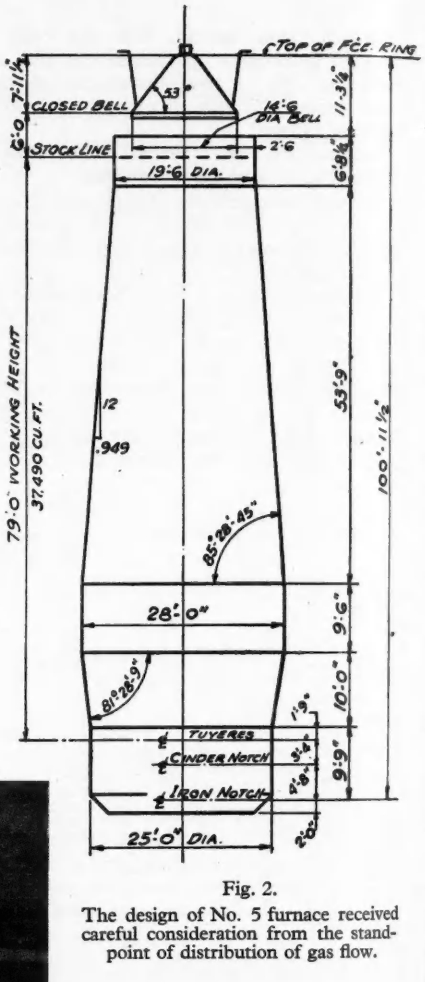


Fig. 2.

The design of No. 5 furnace received careful consideration from the standpoint of distribution of gas flow.

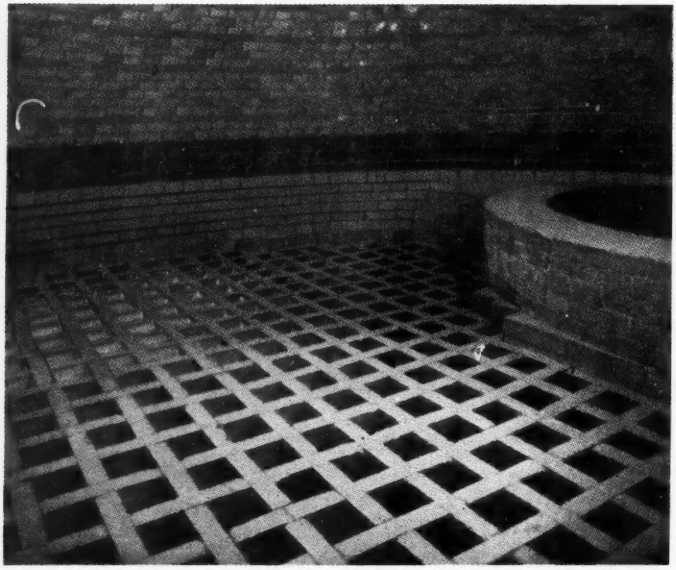


Fig. 3.

Stoves of basket weave construction, with $2\frac{1}{2}$ in. thick brick and $5\frac{1}{4}$ in. openings, have operated 31 months without damage.

All blast furnaces are equipped with a mixer arrangement so that cold air from the engine room can be mixed with hot air from the stoves in definite proportions to attain the required temperature. It has been found that the degree of mixing that takes place varies a great deal, and under extreme conditions there are temperature differentials as high as 500 degrees on opposite sides of the bustle pipe.

We have achieved better mixing by introducing the cold air at the bottom of the hot blast valve drum on each individual stove with automatic temperature control.

Conclusion.

Efficient blast furnace operation is the result of the correct combination of many details. No one detail in itself is of sufficient moment to spell failure or success, yet each is of sufficient importance to warrant careful consideration.

CARBON LININGS FOR BLAST FURNACES

By F. J. Vosburgh. (From *Iron and Steel Engineer*, April, 1940.)

It was recognised that carbon and ceramic materials used for linings of blast furnaces had little in common, and that a different technique was necessary for carbon than for ceramics. It was evident that the usual blast furnace was not designed for carbon linings, for it was not of air-tight construction—a primary requirement if carbon were to be used. The operators found out that carbon blocks could not be used around the tap hole. They discovered that combining carbon and ceramic materials did not work well.

Regardless of the difficulties encountered, it was clear that carbon materials had some distinct advantages. If oxidation could be prevented, carbon lasted remarkably well, for it was attacked by neither the metal nor the slag. If the blocks were well made little shrinkage took place because of the operating temperatures. That carbon did not soften, or lose strength while in use, and that it had a low coefficient of expansion, were good characteristics.

Carbon materials thirty years ago were not comparable with similar articles to-day: electrodes, for example, or brushes, and the carbon brick of that period would not compare in quality with those manufactured to-day.

As a result, developments ran to linings of two types in so far as size of pieces were concerned, the one using standard brick sizes and the other using blocks from large brick size on up to pieces weighing 3,000 lbs. or more.

One of the larger steel companies tried patching a carbon lining with a carbon paste tamped into place. The experiment was successful, and a complete carbon lining was installed in the same manner.

This last type of lining is less expensive than either of the other two (standard bricks and blocks), but is relatively new and not well proved.

According to available information, the mass comprises good foundry coke ground to pass a 1 mm. (0.39 in.) screen, mixed with 15-25 per cent. water-free steel works tar, with preference given to the drier mixers. The mass is well mixed in steam heated mixers, and is tamped into place in thin layers by the use of hand or pneumatic rammers. The rammer heads must be kept hot while in use, and the joints between fresh and previously tamped mix must be well tarred.

Regardless of the type of carbon lining—that is brick, block or paste—the lining is protected during the time the furnace is blown in by a layer of fire brick, usually 4 to 5 in. thick.

Perhaps the most interesting fact in connection with the use of carbon linings is that, in the portion of the furnace so lined, there are no cooling plates and all of the cooling is of the spray or shower type, on the outside of the shell. This reduces the total cost of that portion of the lining and cuts down on the maintenance and operating charges.

The questions of lining life and lining cost are the most important. Germany claims one carbon-lined furnace campaign of over three million tons and several others over two million tons in blast. The writer saw two carbon-lined furnaces that have been in service 10 years with one past two millions tons and the other past 1½ million tons. The only repairs had been renewal of the top of the stack.

There is little information available referring to the cost, as it is difficult to translate

foreign cost to American practice, and then it is difficult to compare the qualities of various countries. The tamped in lining is by far the the cheapest type.

While the carbon lining offers no saving in cost installed, it does offer other possible savings. For a new furnace the construction can be simplified, for no special cooling is required. There is also a saving in the lessened maintenance charges and the simplified construction, no salamanders, fewer run-outs and many less worries.

In order that blast furnace operators may have something upon which they can work in comparing the costs of ceramic and carbon linings of a blast furnace, complete drawings have been made, and from them an estimate of the cost of carbon lining a typical blast furnace up to the mantel. A proposed lining to fit a 1000-ton per day furnace having a hearth diameter of 25 ft. could be installed in the Pittsburgh district for not over $75,000 including freight and labour of installation. It is estimated that the carbon materials for such a lining will weigh before machining 1,300,000 lbs., and that the finished lining will weigh 1,010,000 lbs.

The estimated cost should be compared not only with the cost of a brick lining up to the mantel, but to that should be added the cost of the cooling plates and the piping for them.

The maintenance of a carbon lined furnace is less than a ceramic lined. The carbon lining will maintain its original thickness, so the chances of loss of metal will be nil. The walls will protect the shell evenly, so there should be no hot spots. There will be no cooling plates and complicated piping to keep running.

The carbon block type is seemingly the most successful of all, and appears to best fulfil the requirements of a trouble-free, long-life lining with minimum maintenance.

IRON CORES FOR PIG IRON CASTINGS.

By F. NAUMANN. (From *Werkstatt und Betrieb*).

IT is suggested to use iron cores up to about 4 in. diameter in order to avoid pipes in castings, as shown for example in Figs. 1 and 2. Should the diameter be more than 4 in., hollow cylindrical cores are recommended.

Fig. 1. Fig. 2. Fig. 3.

However, if the hub of a pulley is rather big, as compared with the body itself, as shown in Fig. 3, the cooling effect of an iron core would be too great, and preferably a usual sand core should be used.

It is further suggested to design the iron cores somewhat tapered, a difference of $^{5}/_{64}$ in. in diameter being sufficient, in order to get them easily out of the casting and so to use the same core repeatedly. The cores should be painted thinly with linseed oil and rolled over in sand, before being used.

Rising pipes should be provided in the same way as for sand cores, especially with a view to give free passage to the gases of the burnt linseed oil.

CONTAINER FOR LIQUID FUELS.

By EUGENIO MIOZZI. (From *Schweizerische Bauzeitung*, No. 16, Vol. CXV, April, 1940).

FIG. 1 shows the container designed with a view to use the least possible amount of metal in the construction. Both the inner and outer container **I** and **G** are made of brickwork and outer tank **I** is filled with water to level **n**. A sidewardly placed small tank **V** is filled too with water, and communicates through tube **t** with inner tank **G**. The fuel is being let in or out of tank **G** through pipe **a**, and it floats on the top of the water in **G**, and a weir in **V** maintaining constant water level in the same, keeps the fuel under constant pressure.

As the water level in the outer container **I** is higher than in **V**, inner tank **G** is always under pressure, and thus no fuel can escape through possible leaks, but only water may get through from the outer to the inner tank. This in its turn manifests itself by an overflow through the weir in **V** and indicates at once the existence of a leakage.

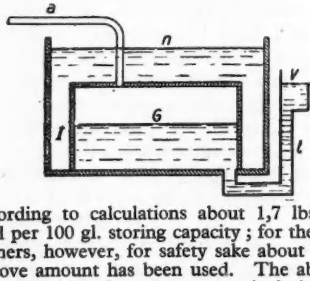


Fig. 1.

According to calculations about 1,7 lbs. iron is needed per 100 gl. storing capacity ; for the first few containers, however, for safety sake about 3,5 times the above amount has been used. The above containers are said to have been extensively built lately in Italy, and it may be mentioned, too, that all are said to have been sunk below the ground level.

CONSIDERATIONS ON THE USE OF HIGH TENSILE STEEL RIVETS IN SHIPBUILDING

By REAR-ADMIRAL W. G. DU BOSE, (C.C.), U.S.N. (From a Paper, presented at the Society of Naval Architects, New York).

EXPERIENCE has shown that fracture of H.T. steel rivets occurs more frequently than in ordinary steel rivets. Fig. 1 is typical, breaking occurring after a time lapse varying from some hours to 2 months after riveting. Sometimes rivets broke during riveting, but very often without apparent cause after riveting. To investigate this, rivets from water tested spaces of torpedo boats were cut out and examined. It was found that a large percentage were cracked where in contact with galvanized parts of plates or frames, Figs. 2 and 3.

Some years ago, a certain builder experienced an abnormal number of broken rivets, this time of ordinary steel, but in a galvanized hull. It was found that the cause was intergranular penetration of the rivets by molten zinc during riveting, and a remedy was found in modifying the technique of riveting.

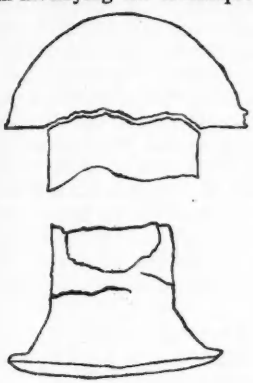


Fig. 1.
Typical fracture H.T. steel rivet from galvanized steel hull.

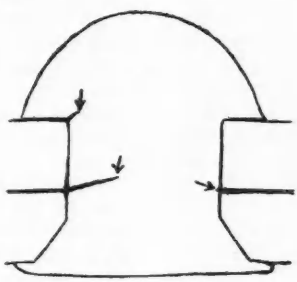


Fig. 2.
¾" H.T. steel rivet from a galvanized steel test piece showing typical cracks.

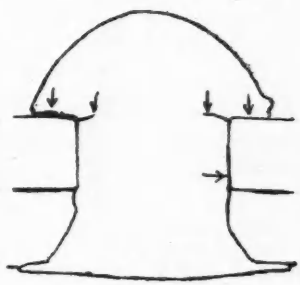


Fig. 3.
¾" H.T. steel rivet from a galvanized steel test piece showing typical cracks.

Tests made to study this subject were carried out on $\frac{5}{8}$" or $\frac{3}{4}$" H.T. rivets taken from torpedo boat hulls with galvanized plating, these were annealed at 1100°F, cleaned in acid and examined by microscope. Of 1297 rivets, 25% were cracked under the head or the point (when not CSK) also at the junction of the two parts connected (but less frequently and only in countersunk rivets); of the cracked rivets, 27% showed large fractures. Also, H.T. steel rivets from non-galvanized plates and ordinary rivets from galvanized plates were examined; the percentages of cracks were 0.4% and 12% respectively, all cracks except one in the latter case, were small.

From the tests it appears that cracks are due to the presence of zinc and are intergranular; the penetration is a well known metallurgical effect, and occurs only if the internal tension and temperature in the metal attacked exceeds certain values. A brittle alloy is then formed, which breaks under the tension and exposes further metal to attack. Neither compression nor sheer stress appears important. Comparing the American practice of galvanizing plates and bars before working with French custom, where galvanizing is often done after working, the latter practice is very bad for the riveting.

Table 1 shows effect of cast zinc on strength of test pieces taken from shanks of ordinary steel rivets.

Test Condition	Tensile Strength lbs./sq. inch	Elongation	Contraction
850°F in air ..	61800	42%	72%
850°F in cast zinc	45500	18%	24%

Experiments were also made to ascertain the effect of variation of the temperature of rivets. In practice the only restriction is that burning rivets shall not be driven, and this temperature varies between 1700°F and 2300°F, according to the type of furnace. Chemical activity increases with rise in temperature; electrically heated rivets were less liable to crack as heating is more uniform and better controlled. Rivets generally were about 10 minutes in the furnace; however, the % of cracks is not greater for rivets heated for 28 seconds in an electric current than for rivets 30 minutes in the electric furnace. Figs. 4 and 5 show that it is desirable

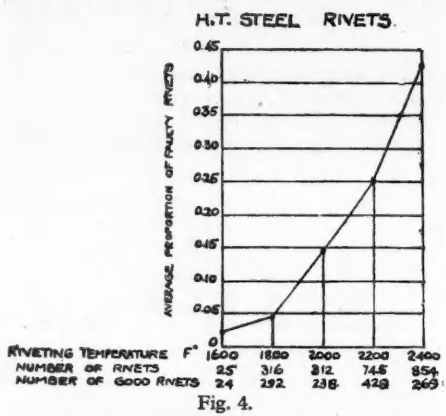


Fig. 4.

to limit temperature of riveting on galvanized plates to 1800°F for H.T. steel rivets and to 2200°F for ordinary rivets. Hydraulic riveting showed much less cracking than with pneumatic work. Further experiments show that manganese content makes little difference to cracking, but since steel with high carbon and manganese content has bad ductility and

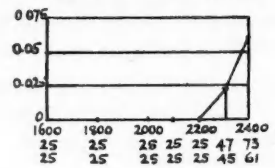


Fig. 5

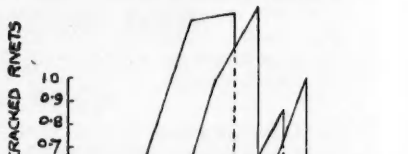


Fig. 6.
Ordinary Rivets.

a plain holes; *b* holes chamfered $\frac{1}{8}$"; *c* as *b* with galvanizing removed rivets with fillets; *d* holes chamfered $\frac{1}{8}$"; *e* as *d* with galvanizing removed.

resilience, it is advisable to limit carbon to 0.3% and manganese to 1.65% for H.T. rivets in galvanized plates. With regard to rivet shape, sharp corners or shoulders with ensuing stress concentrations were improved by fillets under the rivet head and by rounding the edges of the hole, as adopted in the "Lusitania" and "Mauretania" in 1907. A radius of $\frac{1}{8}$" on $\frac{3}{4}$" rivets was adopted, with results shown in Fig. 6, and further improvement was obtained when the galvanizing around the chamfered hole was removed. The shape of the rivet head was found to have little influence, but excessive play of the rivet in its hole about doubled the proportion of cracking. Removal of the galvanizing around the hole could be achieved by countersinking, milling or grinding the plates.

Summarizing, to prevent cracking of H.T. steel rivets in galvanized plates, the following precautions must be taken.

1. The use of rivets with filleted necks in chamfered rivet holes.
2. Removal of the galvanizing from all surfaces in contact with the riveting in highly stressed joints.
3. Riveting temperatures kept below 2200°F.
4. Depth of countersink to be $\frac{3}{4}$ the thickness of the countersunk member.
5. Good practice in respect to accurate erection and tight bolting of the parts is important.

THE HEINKEL COMPOSITE COOLING SYSTEM

(From *L'Aeronautique*, Paris, No. 249, 1940).

ERNST HEINKEL has taken out two French patents on this subject, No. 794,390 and 842,746, the comparison of which is interesting. It is recalled that the Heinkel plane, which holds the world's record for speed, in all probability had a composite cooling system, if one may judge from a photograph, in particular, recently published in *Luftwissen*, showing a small tube protruding from the leading edge, which emitted a light vapour.

In composite cooling systems, cooling by evaporation, the water (or other suitable liquid such as Glycol) discharged from the water jacket of the engine in a liquid state partially evaporates and separates into two circuits. One conducts liquid water to the pump and engine, this water being cooled by the separation of the steam. The other circuit conducts the steam to a condenser, which is actually a radiator cooled by the surrounding air; the vapour loses at this point its evaporation heat.

The principal advantage of the system is that the condenser may be of relatively very small dimensions, hence reduced air resistance.

In such a system the major difficulty is to prevent vapour collecting in the water jacket of the engine, as vapour pockets do not disperse easily from the water jacket and their presence may cause serious local heating.

The Heinkel system—and probably British systems too*—incorporate a valve at the exit of the water from the water jacket to avoid premature evaporation (patent 794,390); thus pressure in the water jackets is increased. The system operates without difficulties near the ground; when cooling is intense, however (at great altitudes or high speeds), the pressure in the condenser falls to very low values, causing a similar drop of pressure in the engine and the undesirable presence of vapour pockets in the water jacket.

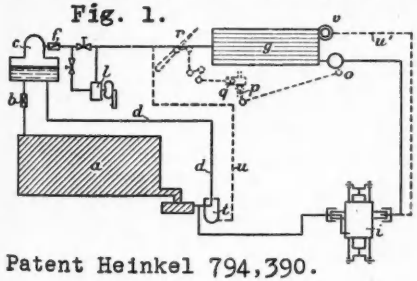
Heinkel has introduced a valve "b" (Fig. 1), fixing in the water jacket "a" of the engine a constant pressure in excess of the atmospheric pressure, and in the evaporator "c" as well as in the condenser "g" a lower pressure than that of the atmosphere. (When starting the engine an auxiliary pump "i" primes the drop in pressure in the condenser which is maintained in operation.)

A second improvement is applied to manual control of pressure and temperature in the liquid circuit to avoid the formation of vapour, whatever the circumstances may be; to this end the pilot uses the valve.

To control the vapour circuit, and especially the pressure in the condenser, the flow of air through the shutter "r" is controlled, for example, by control of the pressure "o" which acts on the oleo-servo-motor "p" "q."

A fourth problem studied by Heinkel is the operation of this arrangement during inverted flight (at a steep angle). In this position the evaporator "c" is replaced by "t," the retaining valve "f" closing up; the new vapour circuit uses "u," "u¹" and the feeding pump "v," while the pump "i" is fitted so that it can still operate.

Fig. 1.



Patent Heinkel 794,390.

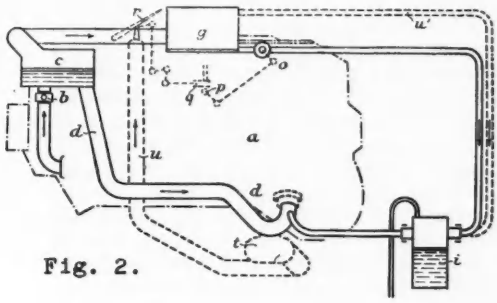
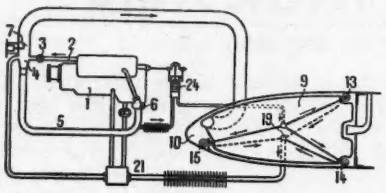


Fig. 2.

Fig. 3.



Patent Heinkel 842,746.

A compressor "1" is provided to hasten the condensation of the steam should the condenser become damaged in combat, or when a considerable strain of short duration is required of the engine, for instance, when climbing or flying at record speed.

Fig. 2. represents an arrangement with the same reference letters as above, but here the evaporator "c" is in front of, and the small condenser "g" above the engine.

Three years later, in patent No. 842,746, Heinkel applied himself once more to this problem.

Primarily, the condenser was maintained at the pressure of the external air, which avoids deterioration in operation if there is a leakage due to bullets; with the above-described arrangement the pressure of the condenser was below the atmospheric pressure.

The valve arrangement which was energised by the water pump of the engine, possibly by manual action of the pilot, is not sufficient to control the system. The temperature of the external air as well as its pressure—which vary according to altitude—have influence on the condenser, at least at speeds which affect the dispersal of calories from the condenser. Thus the system is unstable. If the condenser pressure be reduced, the pressure of the evaporator and of the water jacket are also diminished, and a drop in saturation—temperature—occurs. Thus the water not evaporated returns to the engine, which is cooled above the desired extent.

The new Heinkel arrangement consists of controlling the average temperature of the water in the water jacket by control of the drop in pressure between the evaporator and the condenser.

Fig. 4.

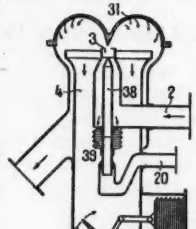
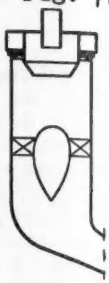


Fig. 6.

Fig. 7.



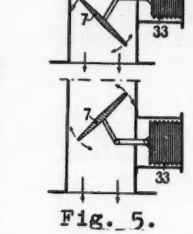


Fig. 5.

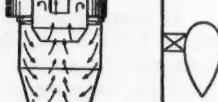


Fig. 8.

Description of this arrangement :—

Fig. 3. Engine "1" where water is at a pressure of 1,5 kg/cm² and at 110°C. The jet "3" brings the pressure up 0,2 kg/cm² at 103°C. Water then passes into the evaporator "4"; "5" is the circuit for return of the liquid water to the pump "6" of the engine. Steam passes through the throttle valve "7," where it takes the pressure of the condenser "9" which is the exterior pressure, so that the pressure remains constant in the evaporator "4."

The condenser is partly formed by the external portion of the wing. Steam first occurs at "10" on the leading edge where there is a tendency to freeze; the surface condenser is thus maintained in a uniform condition which facilitates the control of the mean temperature of the water in the water jacket. References "13," "14," "15" correspond to ejectors which lead the condensed water collected in "19" on to evaporator "4." "21" is an oil cooler. Pump "24" energises the ejectors at a pressure of 4 kg/cm². **

Such is the general scheme.

Notes of the Editor :—

* Early but definite results of British experiments with the "Evaporative Cooling System" are described in *Aero Engines*, Vol. I., by J. D. Frier, published in 1933 by Charles Griffin & Co., Ltd., W.C.2.

** *Aero Engines*, Vol. II. (by various authors), 1939, describing British systems incorporating ejectors, same publishers as above.

Fig. 4 shows the evaporator "4" in detail. Water enters at "2," reduces pressure at the outlet from the jet "3," acquiring speed accordingly. The change of direction produced in the containers "31" is favourable to the separation of vapour towards the centre, and of the water towards the periphery. In "7" the throttle valve is seen, operating with the diaphragm "33" which maintains constant pressure in evaporator "4." The section of the jet "3" is made to be variable by the needle "38" controlled by the diaphragm, fixed in tube "2." When the pressure is reduced in the water jacket, the needle reduces the section of the jet. Needle "38" (being a tube) receives by "20" the water coming from the condenser, which, in its turn is subjected to separation of vapour in "31."

Fig. 5 shows a variant in which the diaphragm "33" is influenced by the external pressure and intervenes more rapidly than in the arrangement described above.

Fig. 6 shows a new arrangement of the evaporator, with a great number of jets arranged, as in a turbine. Fig. 7 is another arrangement, with jets in the upper section and increasing sections in the lower part.

Fig. 8 represents one of the ejectors which receive the condensed water. Water under pressure enters on the right, passes out on the left, sucking up condensed water below. There are three ejectors for the different flight-conditions. Not submerged ejectors—according to inclination of the aeroplane—operate as air pumps.

Finally, the designer has provided an external condenser for the case of maximum power output.

INCREASE OF EFFICIENCY ON AIRLINES BY THE USE OF HIGH-OCTANE FUELS.

By A. VERDURAND. (From *l'Aeronautique* [Paris], No. 294, February, 1940, pp. 68-69).*

IN determining the increase of efficiency on airlines, D. P. Barnard, of The Standard Oil Company, calculates the cost paid per unit volume of fuel as well as the corresponding return. Both the cost and the return increase if a fuel of higher octane number be used. The advantage varies according to operating conditions, but the calculation shows the economical advantage gained by the use of high octane fuels. It is about the same per unit volume of fuel for all modern aircraft whatever the all-up weights, provided similar engines are used, speeds are of the same order, and the rate of the useful load to the all-up weight is the same. The latter ratio is about 35%; the weight per H.P. about 13.2 lb., which corresponds to about 4.4 lb. of useful load per H.P.

The rate of load Cp in lb. transported per American gallon of fuel consumed in flight is given by the formula:

$$Cp = 6 \times \frac{4.5 - \frac{Dc \times fc \times Wc}{Q \times Vc \times Wm} - \frac{C+O}{Wm}}{\frac{Dc \times fc \times Wc}{Q \times Vc \times Wm}}$$

Dc=length of route in miles.
fc=Fuel consumption in lb./H.P.
Vc=Cruising speed m.p.h.
Q=Heat-unit ratio of fuel used to standard aviation petrol /11500 cal./kg./
Wm=Maximum power output, H.P.
Wc=Cruising power, H.P.
C=Weight of equipment in lb.
O=Maximum weight of oil in lb.

In application of this formula to modern aeroplanes such as "Lockheed Electra," "Boeing 247 D." "Douglas D.C.2" and "D.C.3," "Boeing 299," weights from 9,930 lb. to 29,800 lb., the following results were obtained:

1. The economy to be obtained from an increase in rate of compression is about equal to that which can be obtained from increase in take-off power obtained by supercharging. (Let us recall that pushed to the extreme, these two processes exclude each other, unless the engine has variable compression).
2. The economy effected with a fuel of

*Because of the importance of operating costs of transport aeroplanes, attention may also be drawn to a paper entitled, "Some Economic Aspects of Transport Aeroplane Performance" by W. C. Mentzer and Hal E. Nourse, United Airlines Transport Corporation. Presented at the Aeroplane Design Session, Eighth Annual Meeting, I.Ae.S., New York, January 26th, 1940. (Reprinted in the June issue of the "Journal of the R.Ae.S.", London.

high octane number has relative importance all the more considerable as the length of the route is greater and the useful load less.

3. If this calculation be applied to aeroplanes used on inter-American airlines, it is found that the replacement of a fuel of 73 octanes for a fuel with the octane number 87 enables a higher commercial return per unit volume fuel consumed, which return is nearly doubled if one proceed from 87 to a 100 octave fuel.

4. The same calculation applied to Pacific routes shows an even more convincing increase in economy than on shorter land-routes.

THE AUTOMATIC EQUIPMENT IN ANTI-AIRCRAFT DEFENCE.

(From *L'Ingegnere* [Roma], 15th April, 1940, p. 313).

THE rapid development of bombers which have continually increased their speed and operating height has promoted a similar development in anti-aircraft guns. Indeed, the higher the speed of the aeroplane, the greater the error of anti-aircraft fire during bombing in horizontal flight. Actually the cause of errors in estimating firing factors is largely due to altitude. But the increase in power of anti-aircraft equipment as the outcome of increase of calibre and initial speed, makes it necessary for the bomber to release the bombs in horizontal flight with great speed and at high altitude; thus his attacks will be limited to broad objectives (cities, military ports, etc.), whereas accurate attack on small objectives (electric power stations, bridges, battleships, troops, etc.) requires different methods. Thus, when diving and dropping bombs at lower height it is possible to obtain greater accuracy against more compact objectives. Bombing in close formation is the best method against troops on skis, transport columns or lorries.

The success of this method of bombing necessitates a specialized defence, the artillery of greater calibre which is suitable against planes at great altitude being no longer effective. Aeroplanes might navigate in danger zones by zig-zag flight. Surprise in such a way cannot be signalled, except at very short range, when artillery may not have time to intervene. Therefore, in such cases it is necessary to resort to means for defence with weapons which can take aim and fire automatically, with rapid action (150 shots per minute).

But firing at low flying aircraft cannot be effective, except when the aeroplane is at sufficient distance and when the defence equipment is powerful and numerous. Three types have formed the basis of experiments: a heavy machine gun of 13 mm. calibre for bullets of 50g, an automatic small cannon of 20-25 mm. per explosive percussion, and a similar automatic small cannon of 37-40 mm. The heavy machine gun of 13 mm. calibre offers a complete solution of the ballistic problems but the bullet is not effective as an explosive projectile. Even the explosive projectile of 20 mm. is not sufficiently effective to-day, and is insufficient from the point of view of ballistics. Better results have been already achieved by launching a projectile of 25 mm. calibre with double the weight of that of 20 mm. calibre.

CONTRIBUTION TO THE MEASUREMENT OF STAGNATION PRESSURE.

By H. MUTTRAY. (From *Luftfahrtforschung*, Munich, No. 3, 1940).

DESCRIPTION of two new methods suggested for the measurement of stagnation pressure. One of them which is executed on the ground requires the use of optical apparatus. By imitating flight in two directions at right angles, measuring also drift by wind, a time-saving operation is being introduced, avoiding flight in opposite direction. The second system, with indicator in the panel, employs a Pitot-tube for the static pressure, held at some distance from the highest point of the tail plane of usual monoplanes, in a zone where, as is shown, there is no appreciable disturbance of the field of pressure. The method is based on the measurement of the stagnation pressure as difference between the undisturbed static pressure "g" and the disturbed static pressure "p" in the immediate neighbourhood of the aeroplane. These methods present the advantage of immediate application without any preceding flights, adjusting and checking.

THE NEW OLDSMOBILE HYDRA-MATIC TRANSMISSION

(From "*Le Genie Civil*," Vol. CXVI, No. 17, April, 1940).

AMONG the more important innovations in transmission, we may mention the new automatic Oldsmobile Hydra-Matic transmission which combines a four speed automatic geared transmission with a hydraulic arrangement usually known as a fluid fly-wheel (Fig. 1). The transmission comprises in essential two planetary gear sets in series, at the rear of which is the mechanism for the reverse operation. This fluid fly-wheel acts as a clutch and a shock absorber between the engine and the gear box. The control of operation is entirely effected by the accelerator and the brake, and no separate clutch pedal is required.

The fluid fly-wheel consists of two similar rotors, one connected to the engine and the other to the transmission. These rotors are placed face to face and oil is supplied from the transmission at a pressure of approx. 2 kg cm² (30 lbs/sq. in.). The mating faces of the rotors consist of a series of specially shaped passages, and relative movement between the rotors causes the oil to be circulated in a spiral direction as shown by the arrows in Fig. 1. By this means the power of the engine is transferred to the transmission very smoothly. The planetary gear sets are placed in series as shown in the illustration so that power may be transmitted through either or both to produce either of four forward ratios or reverse.

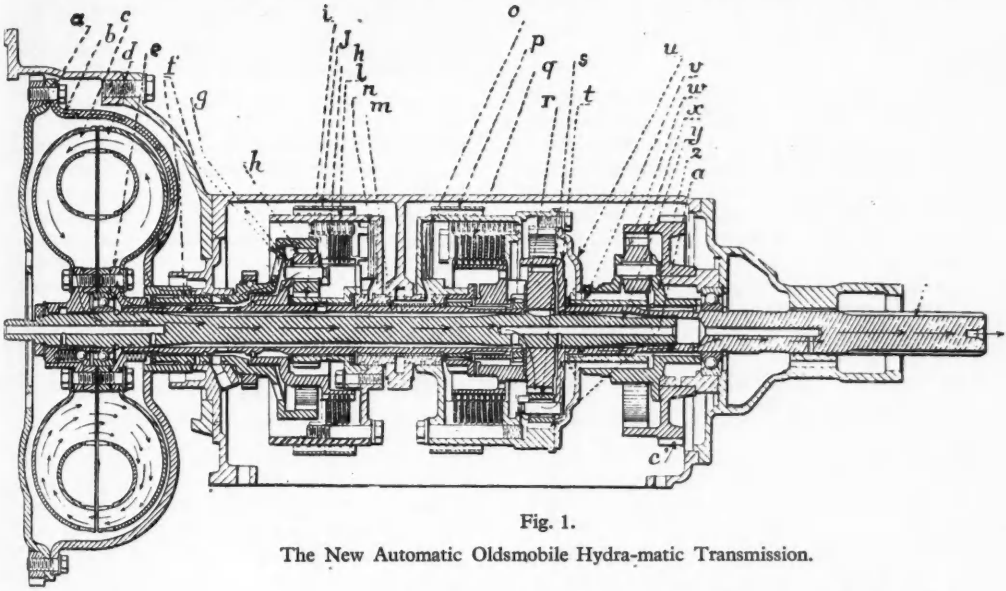


Fig. 1.

The New Automatic Oldsmobile Hydra-matic Transmission.

(*a*) Flywheel
(*b*) Fluid Flywheel Casing
(*c*) Front Rotor (Driven)
(*d*) Rear Rotor (Driving)
(*e*) Rotor Hub
(*f*) Front Unit Drive Gear
(*g*) Front Unit Planet Carrier
(*h*) Front Unit Planet Pinion gear
(*i*) Front Unit Sun Gear
(*j*) Front Unit Brake Band
(*k*) Front Unit Clutch
(*l*) Front Unit Brake Drum
(*m*) Intermediate Shaft
(*n*) Front Unit Clutch Piston
(*o*) Rear Unit Clutch Piston
(*p*) Rear Unit Brake Band
(*q*) Rear Unit Clutch
(*r*) Rear Unit Sun Gear
(*s*) Rear Unit Planet Pinion Gear
(*t*) Rear Unit Planet Pinion Carrier
(*u*) Reverse Drive Flange
(*v*) Reverse Unit Sun Gear
(*w*) Reverse Planet Pinion Gear
(*x*) Reverse Internal Gear
(*y*) Main Shaft
(*z*) Reverse Planet Pinion Carrier
(*a**) Rear Unit Internal Gear
(*b**) Output Shaft

A centrifugal regulator forms part of the mechanism of the transmission and determines automatically the gearing corresponding to the desired speed and torque. The change from one gear to the next is effected by the aid of pistons under hydraulic control, the action of which is reinforced, in certain cases, by springs which actuate the brake bands of the planetary gears and of the clutch.

The speed at which the shifts occur is governed by engine throttle position as well as by the centrifugal governor so that, as the throttle is opened, the gears automatically shift at higher and higher speeds.

Fig. 2 shows how the change of speed is controlled by means of a handle under the steering wheel on the new Cadillac and La Salle models.

The new cars no longer have a gear control lever obstructing the front seats; the " finger tip " control is situated under the steering wheel.

The Ford 60 car is equipped with a semi-centrifugal clutch. Cadillac and La Salle also have changed over to the lighter semi-centrifugal clutch to facilitate clutch engagement. To eliminate the effect of engine movement upon clutch engagement, the linkage between the clutch pedal and the clutch housing has been modified on the Ford.

This new relay linkage consists of a lever and shaft and link intermediate between the pedal and clutch release shaft. The inner end of the intermediate lever shaft is designed so that it acts as a universal joint between this shaft and the clutch release shaft.

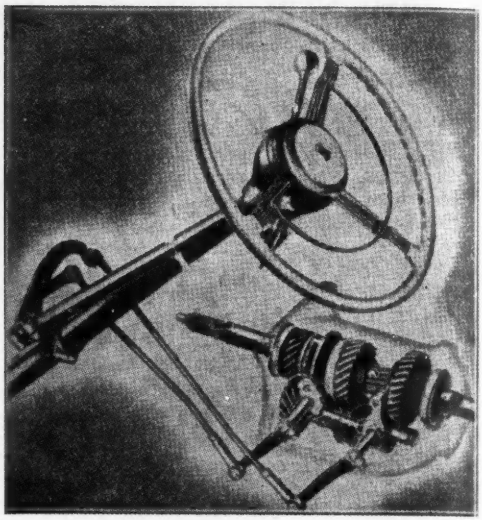


Fig. 2

Pressed steel clutch pedals replace drop forged pedals on the Packards and Chevrolets.

The contact surface of the Chevrolet clutch has been increased by 9%. The pressure on the pedal has been slightly reduced by lowering the rate of the " pull back " spring.

AUTOMATIC CHANGE GEAR FOR CARS

By Dipl. Ing. Hellmuth Droscha. (From *Deutsche Motor-Zeitschrift*, No. 9, 1939).

The **Kreis** change gear consists of two parts: a set of cog wheels being in constant grip and running in an oil bath, and a set of centrifugal clutches isolated from the wheels. Fig. 1. shows the gears in the 1st, 2nd respectively in the 3rd speed, the arrows indicating the way the torque is transmitted. Wheels Z_{1b}, Z_{2b} and Z_{3a} are equipped to this end with by-pass devices, as shown in Fig. 2.

The engine's crankshaft is coupled with

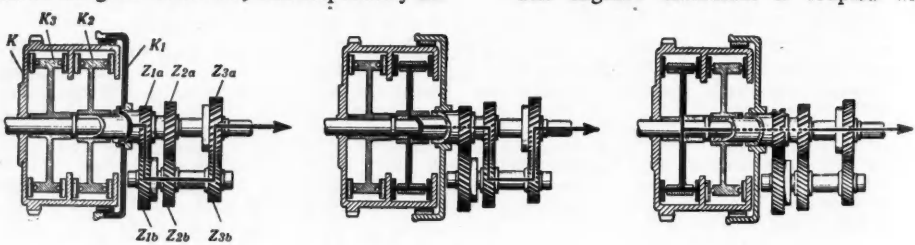


Fig. 1

drum K_1, which is provided with a number of evenly placed centrifugally operated clutch jaws, as shown in Fig. 3. At and after a certain speed these centrifugal jaws come to grips with drum K_1, fixed on a hollow shaft, which bears at its other end wheel Z_{1a}. Thus for the 1st speed the torque is transmitted through gears Z_{1a}, Z_{1b}, Z_{3b}, Z_{3a}, to the main shaft.

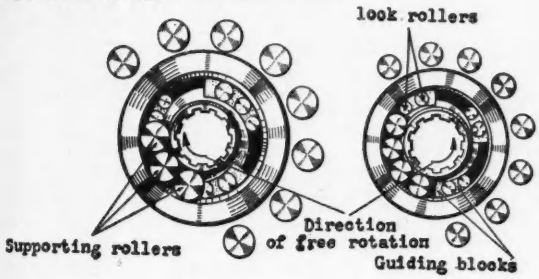


Fig. 2

K_3 comes into operation, this drum being directly fixed on the main shaft, which leads through the two afore-mentioned hollow shafts to the Cardan shaft The centrifugal

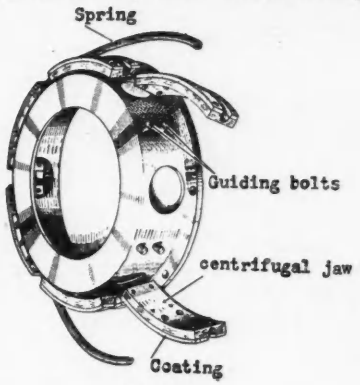


Fig. 3

After the car is speeded up to about 13 m.p.h. the axially working centrifugal clutches of drum K_2 will operate, releasing the balls, as shown in Fig. 4. Drum K_2 in its turn is fixed also on a hollow shaft, which bears at its other end wheel Z_{2a}, and is actually inside the shaft of drum K_1. The torque is thus transmitted through wheels Z_{2a}, Z_{2b}, Z_{3a}, Z_{3b}, while wheel Z_{1b} is being trailed owing to its by-pass device.

At and above 26 m.p.h. the clutch of drum

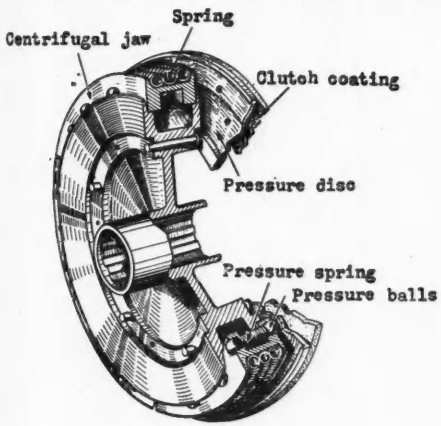


Fig. 4

clutch mounted on drum K_3 is identical with that on drum K_2, but has got stronger springs. In this case the torque is transmitted straight through the main to the Cardan shaft, while wheels Z_{1b}, Z_{2b} and Z_{3a} are running free.

It may be mentioned also that during the 1st and 2nd speed, drums K_2 and K_3 respectively drum K_3 alone, are trailed and rotate at a lower speed than drum **K**. All the three clutches are constructed further so as to disconnect at a considerably lower speed than at which the coupling comes into operation. Thus any backwards and forwards wavering at certain speeds is safely avoided.

Fig. 5 shows a cross section of the **Kreis**

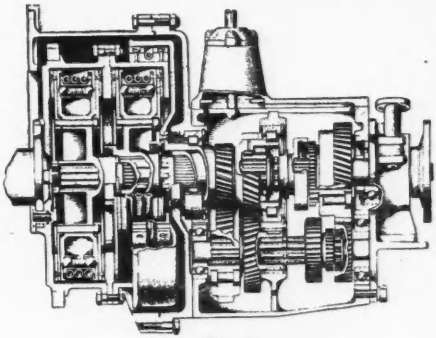


Fig. 5

change gear, showing also the handle which serves to put the whole gear into neutral position; to reverse and to stop the by-pass device of wheel Z_{3a} working when the car runs downhill and the 2nd speed is needed for braking. In order to stop the car running backwards while climbing, a fourth by-pass device is mounted on the hollow shaft of drum K_2, and this by-pass device operates in the opposite direction, as compared with the others.

The afore-described change gear is claimed to weigh 15% more than a usual one, and the costs of production are said to be only slightly higher.

SPRING STARTER FOR I.C. ENGINES

By DIPL. ING. CURT WALLACH. (From *Deutsche Motor-Zeitschrift*, No. 6, 1939).

THE **Volvex**-Starter is illustrated in Fig. 1, the engine being coupled to the starter by clutch socket **1**, which fits into the sliding peg **2**. Casing **3** contains spring **4** attached to spring drum **5**, which mounts spur gear **6**. This may be wound up by gear **7** which is actuated by a handle and secured by catch **12**. The outer end of spring **4** is fixed to drum **5** and the inner end to hub **8**, which has also a brake drum **9**. This brake drum may be arrested by lever **10** actuating excentric bolt **11**.

In order to start the engine, spring **4** is first wound up by hand while hub **8** is held by brake **9** and peg **2** is connected with socket **1**. On releasing the brake with handle **10** the full force of the spring is at once brought to bear on the engine. After the engine has started to work, peg **2** slides back and the coupling between the starter and the engine is interrupted.

This starter is particularly recommended for colonies, where it is difficult to keep accumulator batteries in good order and to recharge them anywhere. Another scope for the Volvex Starter is outboard engines, for which a starter weighing about 12 lbs. is suggested. For heavier engines the starter is stated to weigh 1 to 2 cwts.

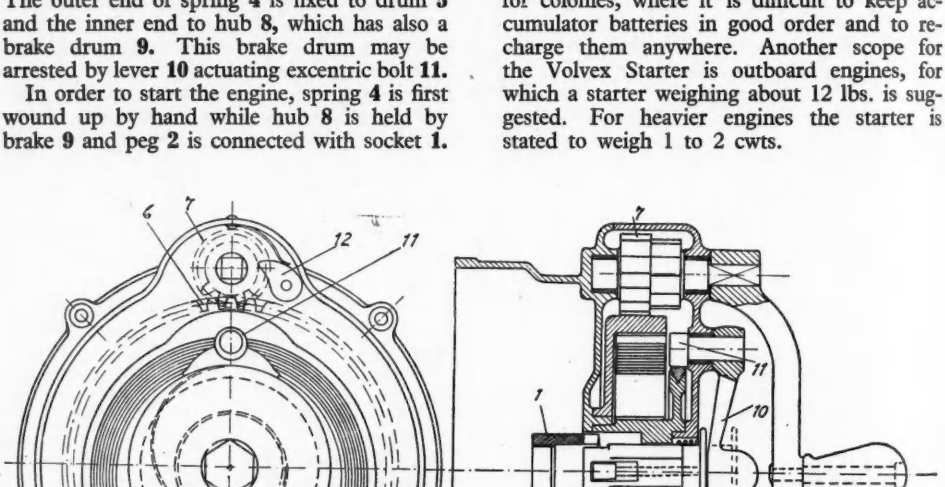


Fig. 1

FLOW PHENOMENA IN HYDRAULIC BUTTERFLY VALVES SUBJECTED TO CAVITATION.

By H. BLEULER. (From *Escher Wyss Nachrichten*, Vol. XII., No. 1-2).

THE forces and torques which occur when closing a butterfly valve may be determined only very approximately by theoretical means. In other words problems are encountered in each particular case, which may be satisfactorily solved with the aid of a suitable testing method. The aero-dynamic test method as employed by Messrs. Escher Wyss is claimed to comply with this requirement, and forms the basis for the design of the new butterfly valves as patented by Messrs. Escher Wyss.

The purpose of the special kind of tests dealt with herein was an investigation of butterfly valves operating under conditions involving cavitation in which case limits are set to the application of the aero-dynamic method.

With regard to butterfly valves subjected to cavitation, two different phenomena have primarily to be considered.

The *first* relates to the actual setting-in of cavitation on the valve. This will only occur to the full extent when the whole hydraulic system on the pressure side, but particularly on the suction side of the butterfly valve, excludes any access of air. At points where the evaporation pressure of the flowing medium is attained, cavities filled with vapour are thus formed.

The *second* case is characterised by the formation of air pockets in the " dead water " regions, which occur at certain depressions and with ideal aerating conditions.

The actual working conditions lie between these two possibilities, i.e., cavities filled with vapour or air occur. In the " dead water " regions air will collect, whilst at the points where the flow adheres to the valve body, for instance at the edge of the same, cavitation may occur. Usually, however, the amount of cavities filled with air exceeds that filled with vapour, since in the case of butterfly valves, which are subject to the danger of cavitation, a fair amount of air will always penetrate through the valves and pits. The formation of air pockets on the back of the valve disc is not unwelcome, for it reduces the torque. There is, however, no fundamental difference between the magnitude of the torque, reaction and flow, when the cavities are filled with air which has penetrated, or with vapour formed from the medium itself owing to cavitation. In the following general remarks, the above-mentioned differentiation may therefore be disregarded.

The testing plant is illustrated diagrammatically on Fig. 1. The motive force is

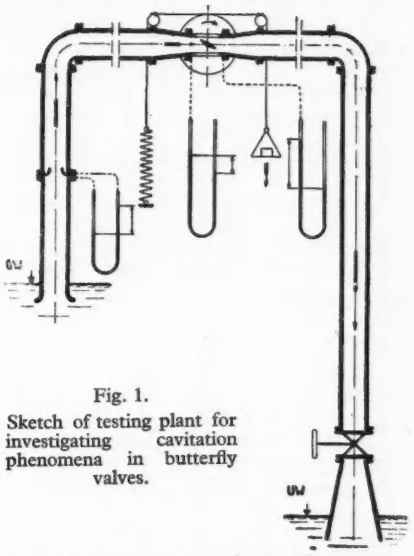


Fig. 1.
Sketch of testing plant for investigating cavitation phenomena in butterfly valves.

obtained from the head in the low pressure test plant. By arranging the observation chamber (cylindrical glass tubes) for the butterfly valve at a suitable height, the natural suction head resulting therefrom provides the small back pressure which is necessary to initiate cavitation. A throttle valve on the suction side serves to regulate this back pressure, whilst the diaphragm is fitted in the admission pipe for measuring the quantity of the flowing medium. Pressure measuring apertures before and after the butterfly valve, together with a device for measuring the reaction of the valve gate, complete the testing equipment.

To characterize the working condition, a characteristic number σ_D has been introduced

similar to the well known Thoma's number σ used in connection with water turbines.

The equation is : $\sigma_D = \frac{H_2 - H_u}{H}$

H_2 designates the static pressure at a given point of the valve discharge pipe. Being an over-pressure it is a positive factor. H_u designates the minimum local depression which is necessary for the formation of cavitation. When operating without inlet of air, $H_u = B$, as with normal water temperatures, the steam pressure of the water is approximately—B. For an ideal air inlet $H_u = O$ because cavitation is then possible already with practically negligible depressions. Since in practice the condition for air inlet is mostly not ideal, and because it is also difficult to ascertain the value of the actual depression, for safety's sake it may be assumed $H_u = B$. The equatation for σ_D may thus be written in the simplified form :

$$\sigma_D = \frac{H_2 + B}{H}$$

H designates the head under which the valve operates, i.e., the difference in static pressure before and after the valve, plus the velocity head :

$$H = \Delta H + \frac{c^2}{2_g}$$

The obtained results can now be plotted on a uniform basis as function of σ_D. This is shown for example in Fig. 2 in connection with the flow coefficient K_D and the torque coefficient K, for that closing angle α of the gate at which the torque attains its maximum value during unobstructed flow through the valve. The torque coefficient was found to drop by approximately 50% when cavitation set in to its full extent. A further critical point was also observed, at which a sudden drop in the torque and flow occurs ; it is designated σ_{Dkr}. The results of a number of measurements for a given type of butterfly valve are shown in Figs. 3 to 5. Here, with σ_D as parameter, the flow coefficient K_D, the torque coefficient K and the valve stress coefficient β are illustrated as a function of the closing angle of the gate.

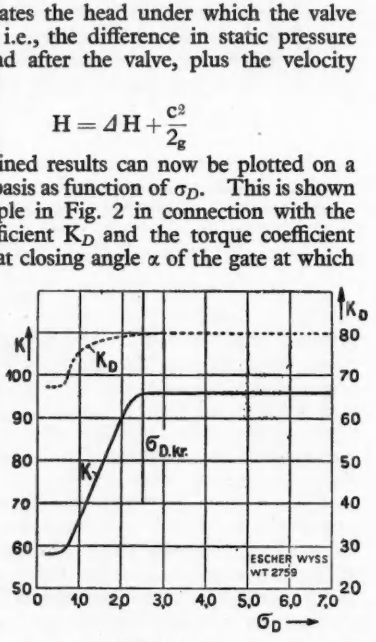


Fig. 2. Coefficient K for the torque and K_D for the flow to be inserted in the formulae :

$M = K \cdot B^3 \cdot H$

$K_D = \frac{C}{V 2_g H}$

$\sigma = \frac{B + H_2}{H}$

M = Torque
D = Diameter
H = Head of valve

B = Barometric level. H_2 = Back pressure.

The coefficients remain constant above the critical figure, whereas below same they drop owing to cavitation phenomena.

With the aid of the above defined value σ_D, the results obtained from experiments can be readily applied to the full-size design, since the model and full-size valve operate under the same conditions when σ_D is equal in both cases.

For the full-size execution, the quantity of water and back pressure which are determined by a turbine, σ_D may be calculated :

$$\sigma_D = \left(\frac{B - H_s}{H_D} + 1\right) \cdot \left(\frac{K^2_D}{K^2_{c\ max}} - K^2_D + 1\right) - 1$$

Herein H_s designates the suction head of the butterfly valve up to tail race level of the installation, and $K_{c\ max}$ the normal velocity coefficient in the cross section of the butterfly valve, i.e., $K_{c\ max} \cdot V 2g$ is that velocity which would result in a cross section of the butterfly valve under a turbine head of 1m., including piping, and when the maximum quantity of water is passing through the turbine.

Observations show the setting-in of cavitation shortly before the critical point, which with falling σ_D then extends over the whole suction side of the inclined valve gate and on both sides of the fully open valve body respectively. Parallel to this phenomenon a sudden reduction of σ_D takes place owing to a drop in the flow coefficient K_D caused by the cavitation. The phenomena occurring below the critical point are therefore unstable and cannot for this reason be easily measured. The same observation has been made in the case of various designs of butterfly valves.

The photographs, Figs. 6 to 10, have not been retouched, and were made with short exposures for operating points below the critical limit. The fully developed cavity on the suction side will be clearly observed. Fig. 7 shows its extension over the whole back of the valve gate, even with a slight closing angle.

For operation in cases where cavitation occurs, i.e. $\sigma = \frac{B + H_2}{H} \leq \sigma Kr$:

Fig. 3

Flow coefficient $K_D = \frac{c}{\sqrt{2gH}}$

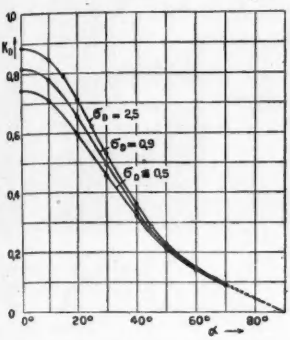


Fig. 4

Torque coefficient $K = \frac{M}{D^3H}$

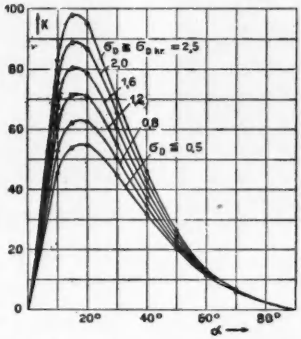


Fig. 5

Stress coefficient β on gate $= \frac{P}{D^3H}$

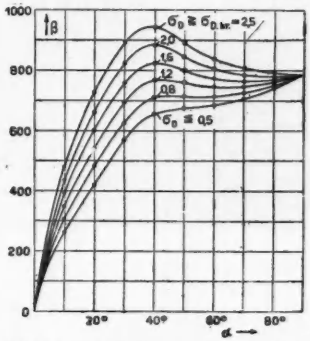


Fig. 6 shows, when compared with Fig. 10, that the cavity due to cavitation is the same with various designs of butterfly valves. Observation of the configurations of the flow made visible owing to the setting in of cavitation in the neighbourhood of the pipes, gives an idea of the complex nature of the phenomenon, and is at the same time a confirmation of what has been said before with regard to the limitations set to theoretical calculation. In any event, Figs. 6, 7 and 10 confirm the formation of marginal eddies, as has to be expected in accordance with the theory of airfoils of finite span. Such eddies have the tendency to rotate around the lateral edges from the pressure side to the suction side, and they will be particularly noticed at a small closing angle of the butterfly valves; but even at a closing angle of 60° (Fig. 8), a rotational movement in the above-mentioned sense will still be noted.

The form of the cavity due to cavitation shows that outside the safety limit the pressure distribution on the suction side is practically constant. Accordingly, for a symmetrical position of the axis of rotation, no movement is set up, which even applies for the position of the gate at which the maximum torque occurs. Thus, it is the pressure on the delivery side only which primarily influences the magnitude of the torque. By neglecting the reaction on the flow passing along the pressure side of the butterfly valve when cavitation occurs, it is therefore possible to calculate, with the aid of pressure measurements carried out by the aerodynamic method, the torque for operating points situated outside the safety limit, i.e., for flow conditions at which cavitation occurs. Particular attention is drawn to the suitability of such measurements in ascertaining the critical point.

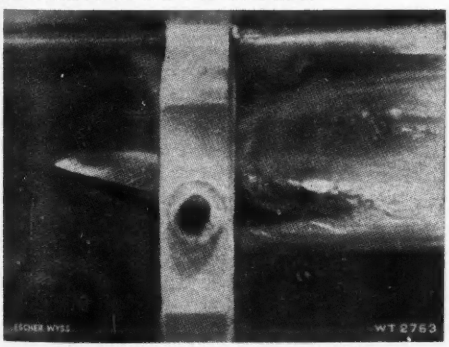


Fig. 6. Closed by 10°. $\sigma_D = 0.186$.

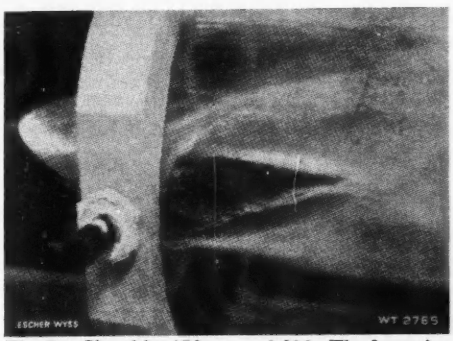


Fig. 7. Closed by 15°. $\sigma_D = 0.306$. The formation of cavitation extends over the rear part of the gate.

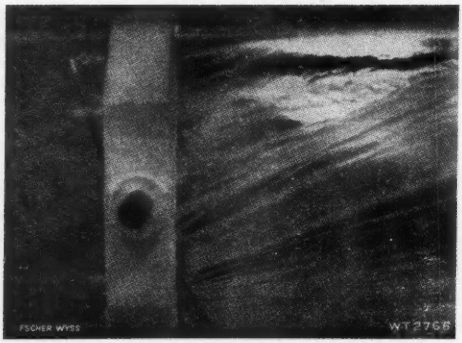


Fig. 8. Closed by 60°. $\sigma_D = 0.560$. The formation of cavitation extends into the whole "dead water" region behind the gate.

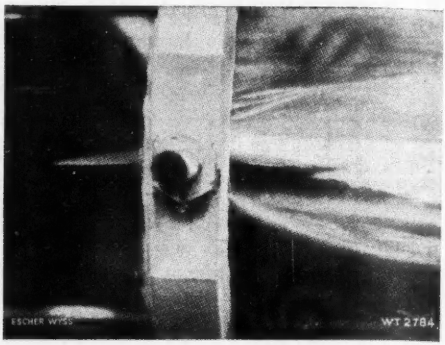


Fig. 9. Fully open. $\sigma_D = 0.2$. Likewise pronounced cavitation phenomena causing a considerable increase in resistance.

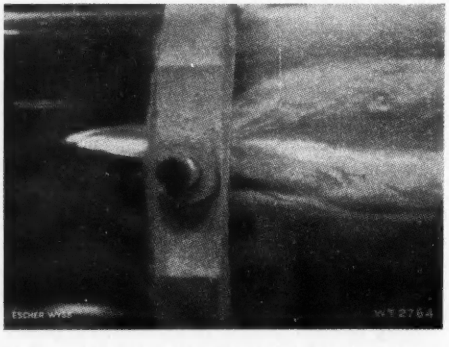


Fig. 10. Closed by 10°. $\sigma_D = 0.186$. Various designs behave in the same manner as may be seen on Fig. 6 showing a cast, lens-shaped gate and on Fig. 10 illustrating a welded design with ribs.

By means of aero-dynamic measurements of the pressure distribution, the following values can be immediately determined:

Minimum pressure .. $p_{min.}$

Back pressure .. .. p_2

Static pressure before valve p_1,

and further the head $p = \frac{\rho}{2} - c^2 + p_1 - p_2$

These values give the dimensionless coefficient

$$\lambda = \frac{p_2 - p_{min.}}{p}$$

Since the laws of similarity apply to the aero-dynamic model and the full-size valve, as long as cavitation does not occur, the factor λ retains its validity also for the full-size piece.

For the full-size hydraulic execution the following designations apply:

$H_{min.}$ = minimum pressure

H_2 = static back pressure

$$H = \text{Head} = \Delta H + \frac{c^2}{2_g}$$

The equation may be written therefore:

$$\lambda = \frac{H_2 - H_{min.}}{H}$$

As already set forth, cavitation sets in when:

$H_{min.} = -H_u$ (H_u being a depression),

or, for example, when operating without air admission, i.e. $H_{min.} = B$,

λ may also be written as:

$\lambda = \frac{H_2 + H_u}{H}$ for general operating conditions, or

$\lambda = \frac{H_2 + B}{H} \sigma_D$ when working without air admission.

This equation corresponds to the one for the cavitation coefficient σ_D when cavitation sets in.

The depression coefficient λ ascertained by the aero-dynamic method is, therefore, equal to the critical σ_D value of hydraulic butterfly valves.

ELECTRICAL MEASURING INSTRUMENTS FOR COLLIERIES WITH DANGER OF FIRE DAMPS

By Dr. Ing. P. M. Pflier, Berlin. (From *Elektrizitaet im Bergbau, No.* 1, 1940).

Danger and protection against sparking.

Sparking may occur if circuits are disconnected, in case of difference between the potentials, or between one part under tension and the other, being earthed.

Protecting measures could be summarised as follows:

1. Limitation of the electrical energy to a value below the ignition limit,
2. or, if this is not possible, safety precautions, both electrically and mechanically must be taken,
3. and if neither of the above-mentioned measures could be applied, the measuring instrument is to be provided with a casing strong enough to withstand the shock of an internal explosion without the risk of any flame reaching the surrounding air of the mine.

The application of any of these protecting measures will vary according to the specific conditions.

1. **Construction with limited output.**

When considering the explosive mixtures as well as their ignition temperatures, then we must determine the limit of the arc energy—quantity which can be admitted in the instrument. It is necessary to find functions $i_z = f(u)$ for each explosive mixture, i_z being the smallest current with a voltage U which may cause ignition under most favourable conditions of mixture. The dependence of this function on temperature and pressure as well as on the shape and material of the igniting electrodes is to be considered. The test results should be accepted only for circuits with ohmic resistance, because the arc energy is increasing in a field containing conductive or capacitive resistance. For various difficulties in the examination whereby the influence of the different factors ought to be ascertained, there has been no publication yet dealing precisely with the whole problem. There are, therefore, two alternatives: either to make experiments in connection with each special case, or to remain far below the ignition limit. An example for the latter method would be instruments containing thermoelectric couples with a tension under 100 mV and a current under 2 mA in a circuit which is nearly without inductance. This will certainly not cause an ignition of the surrounding air-mixture. Even a much higher output can be admitted without hesitation (for instance, as adopted in electrical resistance thermometers with values from 6 to 35 V and 20 mA with a self-induction in the circuit not higher than 0.1H).

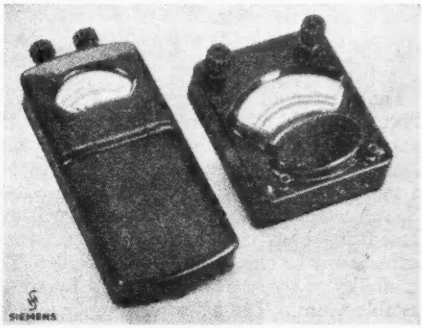


Fig. 1

Fig. 1 shows an example for this type of construction. It is the testing set for resistance measurement produced by Siemens-Halske, A.G. It is a moving coil ohmmeter containing a flash-light battery and an adjustable magnetic shunt. It has several ranges of measurement from 100-5,000 ohms and is embedded into a bakelite case. The casing consists of two chambers, one for the battery, the other for the measuring instrument.

The battery chamber has none but two metal-pieces (terminals), is locked by a special screw and can be sealed with paper. Wiring and connections are laid carefully in respect to short circuit danger.

The chamber for the measuring instrument is supplied with pressed slots for the connecting wires, and there are two small spaces for series resistances wound on porcelain knobs.

Fig. 2 shows the careful layout of the wires

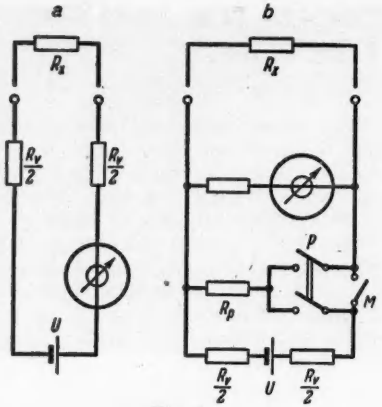


Fig. 2

and the distribution of series resistance on both battery poles.

The tension of the battery is 4.5 V; if a short circuit occurs between the terminals the measuring current will not exceed 22 mA. Should a simultaneous circuit happen in one of the resistance coils, the current will rise to 43 mA; the excessive temperature will not exceed 50° even if the measuring set would permanently be charged with this current. The sensibility of the set can be altered by an adjustable shunt. The adjustment range is 10%, consequently the batteries can be used up to a working tension of 3.6 V.

Briefly: the measuring output and working tension are limited to not dangerous values; short circuits and earth connections are rendered improbable as a consequence of the separate laying of wires, as well as by the sufficiency and the adequate distribution of the insulating material, an unauthorised opening of the casing is prevented by special screws and seals, the working temperature is limited to small values. The ohmmeter has been approved by the Colliery's Testing Establishment in Dortmund.

Fig. 3 demonstrates a smaller testing set, constructed on similar principles.

It contains a bar battery of 1.5 V.

As a supplement for small measuring range an ohmmeter OT1 has been developed, where the unknown resistance RX is not connected in series, but paralleled to the testing set (see Fig. 2b). The testing set has two push buttons. When switch P is closed, a testing resistance Rp is connected with the circuit and the instrument is being adjusted to the utmost deflection by means of the magnetic shunt. Then switch M is closed and the tension is applied to the measuring resistance RX.

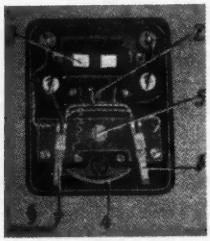


Fig. 4

Fig. 4 shows the back of an instrument with removed cover.

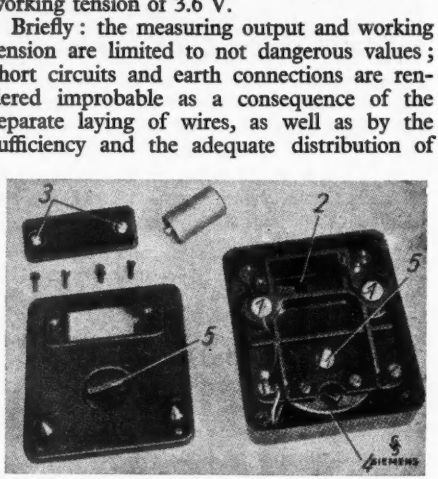


Fig. 3

2. Measuring Instruments with Increased Safety.

Measuring instruments, the output of which cannot be limited to a safe value, have to be made strong and safe, to render impossible any breaking of wires, short circuiting or earth connection. Electrical safety can be obtained by avoiding moveable wires, by small load per wire section, long air and surface leaking paths. The safety is guaranteed by strong casing, small yet thick glasses, strong scale sheets and safe fixing of all parts.

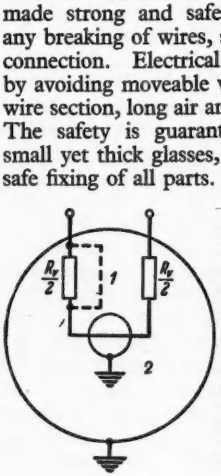


Fig. 5

The series resistance of a voltmeter is divided into several smaller resistances at both ends of the voltmeter winding. These smaller resistances are to be charged to a temperature lower in any case than 40°C. Even when charged permanently with $V\overline{3x}$ tension there must be no dangerous heating. By

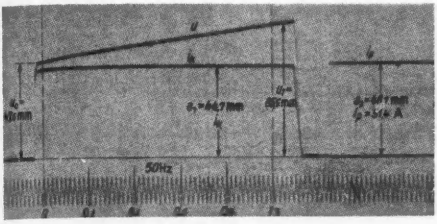


Fig. 6

arranging the resistance at both ends of the field coil (Fig. 5) the safety has been increased.

Amperemeters should have an even greater over-load safety; a 50-fold overload per sec. is required.

Instruments not able to withstand such an overload are supplied with a protective current transformer. The over temperature of the instrument winding is established **for the type test** with the 50-fold current during 1 sec. by means of an oscillograph, and **for the single instrument test** with a permanent charge by the nominal current.

In Fig. 6 an oscillogram is shown, made for a type test.

Fig. 7 is demonstrating the measures which have to be taken to insure the mechanical safety. The casing of the instrument is made of 2 mm. thick sheet-iron. The front end has a bigger diameter and ends in a flange. Between the cover and the flange is placed a joint of round rubber, a circular plate of 5 mm. "Kinon" glass and a second flat rubber joint. The cover itself is made of 4 mm. sheet iron and is fixed by eight countersunk screws, two of which are sealed.

Fig. 7

Precise insulating precautions are observed also at the remainder of the structure. The fittings lead into the terminal chamber, which is made of 2 mm. sheet iron welded to the casing. The cable fittings are provided with rubber joints. All screws are covered with lac and safeguarded by springing washers. The field coils of amperemeters have a great cross-section, the current density is 3.5 A mm^{-2} the over-temperature at the permanent charge less than 15°C and at 50-fold current-load for 1 sec. less than 140°C.

At a nominal tension the series resistance of voltmeters have an over-temperature of less than 30°, the field coil less than 20°.

Each instrument is charged for a certain time and the over-temperature is measured. Between the terminals and the casing a tension of 2,000 V is applied for one minute, and then each instrument is tested as to whether the results of the type test can be applied to the single instruments. The results are marked on certificates issued with the instruments.

The type tests also comprise a long period charge test, shaking test and fall tests.

3. Casing.

A pressure resisting case has to be used if an instrument contains parts which are sparking in service and if it is impossible to guarantee a sufficient safety (for instance, instruments with moving current paths). To this class belong the moving coil instruments, the dynamometers, instruments with a working tension higher than the admissible limit (250 V according to the old and 40 V according to the new prescriptions). Tight casing of instruments should—if possible—be avoided, for they are expensive and difficult to handle.

Fig. 8 shows a device used in connection with a liquid meter for remote counting. The switch is placed in a pressure-resisting case, although sparking cannot occur in service.

The cylindrical part of the casing A contains the parts needed for the connection to the pointer mechanism, it is screwed to the head of the water meter by screws 14. The pointer mechanism of the water meter drives the stud 1 which is coupled to the guiding plate 3 by an interchangeable gearing 2. The plate 3 drives a pin 4. Moved by plate 3 to a new position the pin is brought back to its initial position by springs. A reversible mercury switch 8 is attached to axle 5 of the pin 4. Switch 8 is placed in the second pressure-resistance chamber B; the current impulses

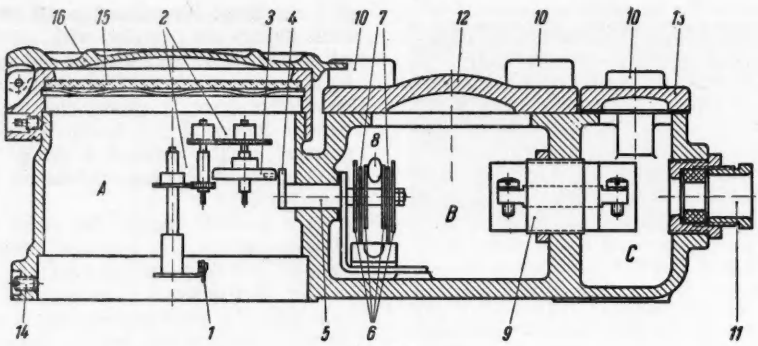


Fig. 8

of the mercury switch operate a remote counting mechanism. The terminals are situated in a third chamber C which is built according to the principles of " increased safety." Thus the cast iron casing is composed of three chambers :

A, which is not connected, because it does not contain electrical parts ;

B, containing the mercury switch is safe against a pressure of 10 at ;

C, with the terminals is built with increased safety.

This terminal chamber C can be closed by a cast iron cover with an intermediate layer of a petrol and oil resisting rubber joint, held in its place by set screws. The terminals are safeguarded against loosening by safety angles and springing washers. The cable is led into the terminal box through a water-proof fitting. The insulation is tested against the casing with a tension of 2,000 V.

Fig. 9 demonstrates the complete instrument in detail.

Measures taken to prevent explosions.

The switch has been built as a mercury reversing switch, therefore sparking could occur after a mechanical destruction only. The terminal box has been built with increased safety, the switch has been placed into a pressure resisting chamber, the type of the construction is stated on a label.

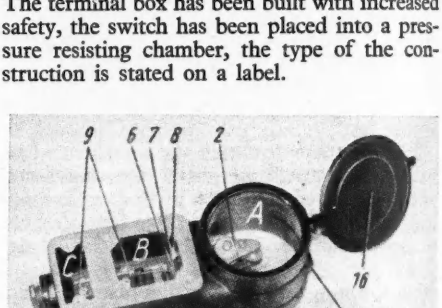


Fig. 9

CRITICAL COOLING RATE OF IRON-CARBON ALLOYS.

IN the June *Journal of Research*, a method is described for heating small specimens of steel in vacuo and in an atmosphere of dry nitrogen to different temperatures and quenching directly in hydrogen. Determinations were made of the austenitic grain size and critical cooling rate of high-purity iron-carbon alloys ranging in carbon from 0.23 to 1.21 per cent. These data show how carbon influences the austenitic grain size and critical cooling rate of the alloys, and make possible a direct comparison of the relation of austenitic grain size to critical cooling rate of hardenability.

THE AIR CONDITIONING PLANT IN THE OFFICE BUILDING OF THE GENERATING CONCERNS IN PRAGUE

(From *Strojnicky Obzor, Prague*, Vol. XIII, No. 16).

An office building recently erected in Prague has been fitted with a modern air conditioning and plenum heating plant, and is described in a paper by Mr. Kletschka.

This building accommodates 1,100 persons, and is designed to secure the most efficient working of the various departments, i.e. the generating concerns, the tramway and bus passenger company, and the district heating company.

Fig. 1 is an illustration of the building described. The rooms vary in size, being approximately 16 ft. × 16 ft. to 16 ft. × 6 ft. (*query height or cubic content*). In addition to the offices, there is a large hall, showrooms, a canteen with electric kitchen, a cinema, certain laboratories, bathrooms and a surgery.

Fig. 1.

The frame of the building is in reinforced concrete filled in with walls of hollow fletton bricks. The windows are large and are fitted with double glass, while internal canvas blinds help to reduce the radiant heat from the sun.

The proximity of a goods station with coal depots, together with the dust and noise produced by heavy passing traffic made mechanic ventilation a necessity, and in the design of the plant the latest American practice and experience were incorporated.

The conditions governing the design of the plant were :—

Outside temperature (Winter)	..	−4°F.) Humidity assumed
Outside temperature (Summer)	..	+86°F.) to average 40%
Internal temperature (Winter)		+68°F. @ 38% A.H.
Internal temperature (Summer)		+77°F. @ 55% A.H.

The limits of tolerance being plus or minus 2.25°F.

The heating and air conditioning plant is divided into 45 different groups which consist of rooms with the same aspect and similar conditions on each floor.

Each of these groups is supplied with fresh air by one of the 21 air conditioning units placed on the different floors. These units consist of a fan with motor, steam heating battery, a temperature control and a steam control valve, a filter, an air mixing chamber and a plenum duct. The conditioned air is conveyed into the rooms via sheet metal ducts, and the extraction of the air for re-circulation first passes an oil filter, and thence to the central air conditioning plant in the basement, where it is mixed with fresh "conditioned" air and re-circulated.

The air interchange can be varied from four to twelve times per hour.

The galvanised sheet metal ducting is incorporated in the building, and an intimation that the weight of this ducting is 150 tons will indicate the size of this system of ducts.

Thermostatic control over a wide range is essential. For winter conditions the thermostats control the steam flow to the heating batteries, and are regulated by the outlet temperature of the air heaters. For summer conditions, the fresh air is reduced to 45°F. in the central air conditioning plant, and the thermostatic control regulates the proportion of this low temperature air mixed with the re-circulated air. Compressed air at 21 lbs/sq.in. is the motive power for the thermo operated valves, etc.

The ventilation for the lavatories, cloak

rooms, shops, stores and surgery is by extracting the same quantity of air as is blown in. For this exhaust ventilation propeller fans are used.

The central air conditioning plant is of standard type and design, and is illustrated in Fig. 2; it consists of:

An air washer.
Two refrigerating compressors.
Air heaters, etc.

The refrigerating plant has a capacity of 3,440,000 B.Th.U.'s per hour. One compressor dealing with 66% of the maximum, and the other 33-1/3% of that load—the condensers are located on the flat roof of the building. The electric motors driving these compressors operate on an A.C. current of 2.850 volts, and are rated at 200 and 100 K.W. respectively.

The primary heat for this plant is obtained from the mains of the District Heating Co. and is received at a pressure of 210 lbs/sq. in. abs., the secondary pressure being 30 lbs/sq. in. abs.

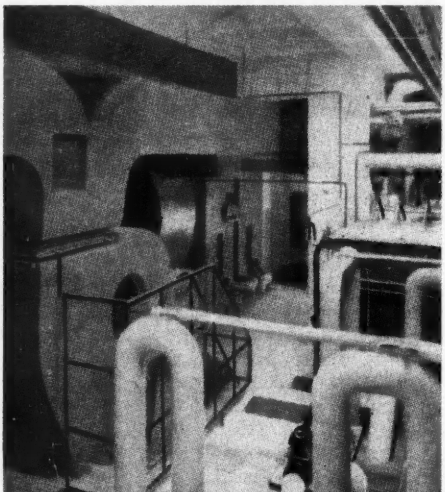


Fig. 2.

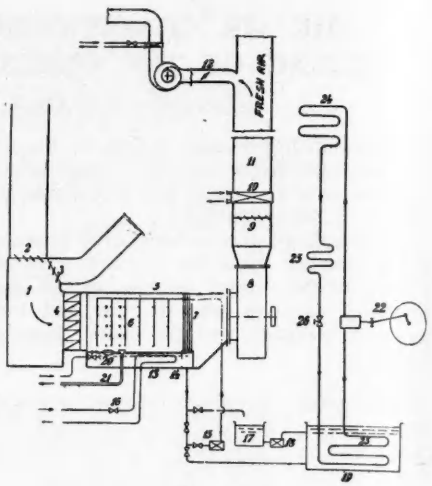


Fig. 3.

Scheme of the Air Conditioning Plant.

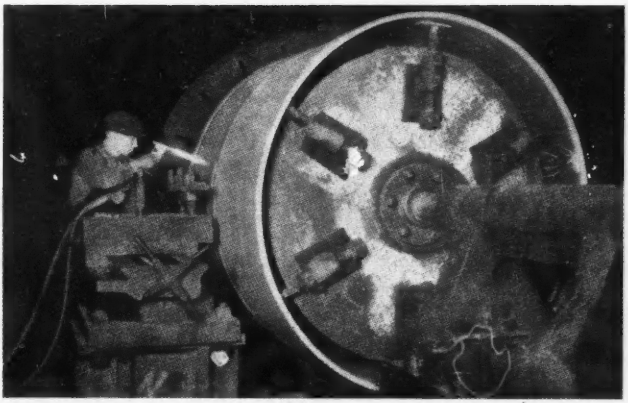
1. Mixing chamber; 2, 3. Control dampers; 4. Oil filter; 5. Air washer; 6. Air humidifier; 7. Dehumidifier; 8. Fan; 9. Control damper; 10. Heating battery; 11. Air duct; 12. Fresh air duct; 13. Tray; 14. Strainer; 15. Pump (to nozzles); 16. Steam control valve; 17. Water tank; 18. Circulating pump; 19. Cooling tank; 20. Ball valve; 21. Overflow; 22. Ammoniac compressor; 23. Cooling coil; 24, 25. Condensers; 26. Reducing valve.

for the domestic hot water supply, and 18 lbs/sq. in. abs. for the comfort heating plant.

The total steam consumption is 7 tons per hour during the winter, of which 12.4 million B.Th.U.'s per hour is absorbed in the heating system, and the remaining 6 millions maintains the H.W.S., etc. The output of the Hot Water Calorifiers for Domestic Hot Water Supply and Hot Water for the Baths is approximately 6,600,000 B.Th.U.'s per hour.

The consumption of steam is metered both as steam and condensate.

The experiences with the above described plant were perfectly satisfactory, and the first regulation of 1,200 air outlets was not difficult at all. The plant is working entirely automatically. During the first heating period, however, it was found that the inside temperature of 68° F which was taken as a basis for the design, was too low, and the temperature of 70° F was required by the greater part of the staff.



A large expansion joint being coated with bronze. The view at the left illustrates the application of silica-zinc to a complicated internal section of a submarine.

A PISTOL THAT SPRAYS POWDERED METALS AND PLASTICS.

By R. G. SKENNETT.

(From *Compressed Air Magazine*, No. 1, Vol. XLV, pp. 6057-6060.

THE Schori spraying pistol has a wide field of use as it makes possible the application of metals and plastics not only to metal, but also to wood, plaster and other substances.

Several European governments have standardised on this process as a means of protecting some of the surfaces of submarines and other war-craft. Ballast tanks are now metallised by this type of equipment, and the same method is employed to protect various valves, lifeboats, and other accessories of sea-going ships.

Schori pistols are used to metal-spray ammunition, submarine mines and the projectiles for large guns. In the latter case, the coating, apart from safeguarding against rust, acts as a lubricant and thus reduces the wear of the rifling.

Recently the process has been proved to be practicable for the spraying of powdered glass on metal surfaces, where the metal is of relatively thin section.

The ultimate industrial solution of this problem, however, is still a matter of research. Success depends upon the discovery of a glass having a low co-efficient of expansion, and also a comparatively low melting point.

What is known as silica-zinc has been used for some time in a powdered form, especially for coating metal surfaces exposed to the weather and to salt spray and moist air from the sea. Silica-zinc is really a mixture of powdered zinc and powdered glass. The melting point of zinc is 780° F. and the fusing point of glass is somewhat higher. By employing a very finely ground glass in combination with the zinc, the Schori pistol will melt, with a suitable mixture of gases, the zinc and the glass and produce a coating which is distinctly superior to that of zinc alone.

The pistol has lately been successfully used to spray coatings of such plastics as hard rubber, thiokol and cardolac. Thiokol is a synthetic rubber, and films of it are described as resistant to oils, greases and many acids and solvents that are destructive to unprotected metal equipment of various sorts.

It is claimed that pistols of the Schori type

are capable of coating at the rate of 250 sq. ft. p.h., and, because of the durability of metal as compared with paint, the resultant over-all economy gives the metal film distinctive advantages.

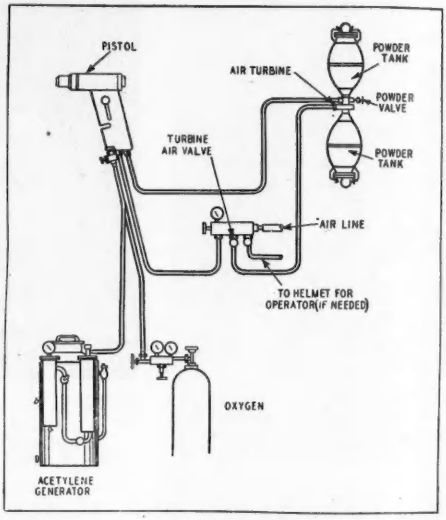
It is a great advantage of the Schori pistol that it has no moving parts that would wear out or need recurrent attention. Only the nozzle where the actual melting takes place, is subject to wear, and an impaired nozzle may be removed and a new one screwed on in a few moments.

The Schori pistol is operated by compressed air so brought into action that it draws the powdered material by an induced suction into the tool, then impels the suspended particles to the zone of the melting flame, and imparts to the molten globules a high velocity that will carry them with an impacting force to the surface which is to be coated. The gases for the melting flame enter at the bottom of the pistol grip and are discharged through concentric annular outlets at the nozzle, where they create a fusing zone through which the powdered materials have to pass.

The temperature range of the melting flame can be varied either by regulating the feed of the given gases or by the use of combinations of gases that have greater or lesser maximum temperatures when ignited. Because of this variable control, the Schori pistol can be adapted to meet the respective melting points of different metals or of any of the other materials that may be sprayed with it.

Whenever different metals or other substances are sprayed successively by the same pistol, both the air and gas pressures are adjusted for each one in turn to give the best operating conditions.

The powdered metal used may be either a direct product or a by-product of a manufacturing operation; and, in either case, because of its source and form, it can commonly be marketed at a lower price than wire of a prescribed gauge and of like metal. Powdered metal of any desired grain size and specified purity is obtainable in ample quantities to meet the requirements of the new form of spraying pistol. Oxides of the given metal can be admixed with the powdered metal with satisfactory results, because the heat of the melting flame is high enough to reduce the oxides and to combine them with the metallic particles,



The drawing shows the various working parts of the Schori system and indicates their relationships. Either half of the powder chamber may be swung into top position so that its contents will flow by gravity into the feed line to the pistol.

so as to produce a solid and uniform film when sprayed. Obviously, if the oxides were not reduced and converted to a true metallic state, then the film would be a spongy and imperfect one.

The flame used by the Schori pistols may be the result of combinations of oxygen, acetylene, butane, propane, or some enriched form of domestic gas. Combinations of oxygen and propane afford a flame of ample heat to deal with numerous kinds of brasses and bronzes, as well as with many modern plastics, when reduced to a powdered form. On the other hand, oxygen and acetylene give a flame of much greater heat and are, therefore, capable of fusing materials that have notable high melting points. Although tungsten has a melting point of 6,100° F., it has been successfully sprayed after passing through a pistol flame having a lower temperature than melting point. The explanation is that the flame is hot enough to make the tungsten plastic, and when the metal is so softened it can be sprayed and deposited by an action that is seemingly akin to welding.

In the earlier forms of spraying pistols that were designed to handle powdered metals, difficulty was encountered in maintaining a continuous flow of the material from an exterior reservoir to the pistol. The powder tended to pack recurrently in the flexible tubing of the feed line, or otherwise to obstruct the desired movement. This occurred, it is reported, even though the powder was agitated by means of compressed air discharged into the reservoir. F. Schori has overcome this difficulty by devising a double chambered reservoir that has the outward modelling of an hour-glass. At the junction of the necks of the two tanks he has interposed a small, enclosed, air-driven rotor that is mounted on ball bearings and set slightly out of balance. As this horizontal rotor revolves, its eccentricity causes it to transmit to the feeding container vibrations that effectually prevent any " bridging " of the material as it flows by gravity from the tank down into the tube that connects with the pistol. The rate of discharge from the container may be regulated by a valve located between the two chambers. The latter are arranged one above the other, and each can be swung in turn into the upper position and empty itself. The powder reservoir is, in reality, much simpler than the description of it.

One of the pre-requisites to all successful metal spraying is that the surface to be coated first be thoroughly cleaned. Sandblasting is usually best for the purpose, and the protecting film should then be deposited with the least practicable delay before the surface is affected by the atmosphere, or anything else that will interfere with the close adhesion of the sprayed film. It is equally desirable that the air used in the pistol shall be clean and dry. Therefore, a properly equipped plant for the work should include apparatus for drying and cleaning, and these units are interposed between compressor plant, the powder reservoirs, and the pistols. The operating air has an average pressure of 50 lbs./sq. in. at the receivers.

MODERN HIGH SPEED LATHE WITH ELECTRICAL COPYING DEVICE.

(From *Das Eisenbahnwerk, April*, 1940).

THE design of the high speed lathe, a description of which follows, represents an important development in economical machine tools for the workshops of all engineering industries. The outstanding point of the machine is a newly developed electrical feeding system for copying profiles without sudden steps.

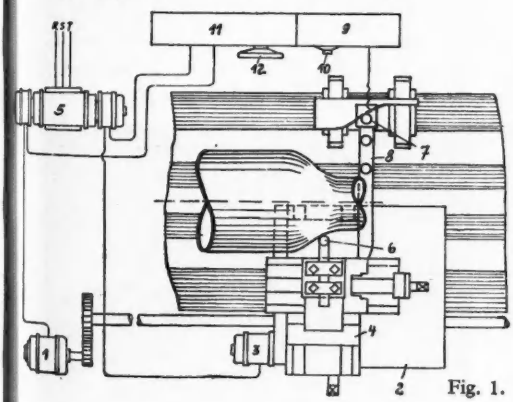


Fig. 1.

As for the mechanical part of the design, the most important features may be described in brief. The machine has four slides and distinguishes itself by its high cutting performance. It is driven by an A.C. motor. The speed of the spindle is varied by gear wheels on a splined shaft so that only the wheels actually working are in gear at any moment. There are 16 possible speeds, thus enabling any job to be done in the most economical way at the highest permissible speed. The headstock is completely enclosed and bolted down on to the bed of the machine. Its cover forms an oil container from which the oil drips on to the various gear wheels which are thus continuously lubricated. The oil leaving the wheels is collected in a sump in the bed and is returned to the container in the headstock cover by a pump. The bearings of the headstock spindle are fed by a forced lubricating system with fresh oil. A sight feed glass at the front of the headstock enables the amount of oil used on each bearing to be controlled at will. The lubricating system for the bearings is operated

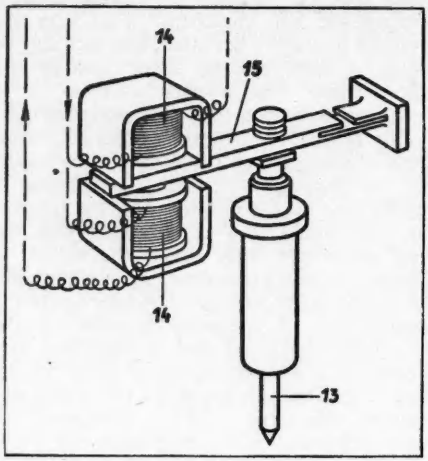


Fig. 2.

mechanically through the spindle, and the machine can only be started when the oil pump for the gear wheel lubrication is working. For ordinary turning, the toolholder, running on the two front slides, can be fed longitudinally and crosswise in the ordinary way, so that all cylindrical and conical turning operations can be performed on this lathe. To the toolholder there is also fitted the above mentioned automatic feeding device for turning according to a master form fixed to the back slide. This automatic feed enables the machine to copy all inside and outside profiles (without sudden steps) in such a way that any desired form can be obtained. This is achieved by two feed motors, one for the longitudinal feed (1, Fig. 1), and one for the cross-feed (3), fixed to the saddle, which can be controlled in a maximum ratio of 1:10 by the speed regulating machine set system "Leonard," so that by varying the speeds of these two motors, the tool (6) is given a motion parallel to the profile of the master. The speed control is actuated by a feeler (7), which is connected rigidly to the cross-slide (4). In this way, the feeler follows the same motion as the tool (6).

The feeler consists of two "Eltas-gauges," one for the longitudinal and one for the cross feed, and is capable of sending out the necessary impulses for the speed regulation with a degree of accuracy of less than a thousandth of an inch. The design of these two "Eltas-gauges" can be seen from Fig. 2. It consists in the main of two coils (14) between which the core (15) moves according to the form to be copied. The two coils are connected across an A.C. bridge. With the smallest movement of the feeler-tip (13) small currents are set up due to variation of the magnetic circuit in the bridge, which, passing through amplifiers, excite the wire gauzes of six "Thyratron" valves and are passed on through the necessary rapid relays to the controls of the motors. According to the position of the cores in the two "Eltas-gauges," the necessary speed and direction of rotation of the two motors is regulated by the "Leonard" controller fixed to the switchboard. This controller is arranged so that the speeds of the motors change in opposite ways in accordance with a sine-curve. By this means the resultant feed motion of the tool is kept constant, so that the depth of the cut remains independent of the slope of the curve. By means of a general control (12) fixed to the "Leonard" part of the switchboard (11), the speed of the two feed motors and the depth of the cutting can be regulated. The whole valve set with the necessary automatic and rapid relays is contained in a completely enclosed part of the switchboard (9), in which there are also additional switches for varying the sensitiveness of the feeler, which can thus be adjusted to suit the type of job in the machine. In addition to this there are various signal lamps and an optical indicator in the headstock, so that the working of the feed motors can be watched and the direction of the motion resulting from the speeds of the two feeds is seen instantaneously. On the feeler there is also an emergency switch which stops the machine immediately if there is anything wrong, so that the job cannot be spoiled. Fig. 3 shows the tool with the feeler and the stencil in operation.

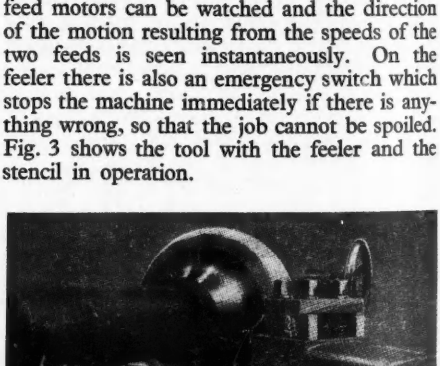


Fig. 3.

The following are the main dimensions of this modern and economical machine tool:

Height of centres	2 ft. 3½ in.
Distance between centres	11 ft. 6 in.
Max. diameter above bed	4 ft. 7 in.
Max. diameter above tool-holder	3 ft. 6½ in.
Chuck diameter	4 ft. 3 in.
Speed of machine, various between	1.5 and 120 r.p.m.
Input to machine	50 H.P.

RECENT DEVELOPMENTS IN WIRELESS COMPONENTS IN FRANCE.

By MICHEL ADAM INGENIEUR, E.S.E. (From *Le Génie Civil*, No. 3, Vol. CXVI).

THE seventh annual exhibition of radio components was held at the Centre Marcel-Berthet, Maison de la Chimie, from 30 January to 1 February, 1940. It was the first exhibition of this kind to be held since the outbreak of war, but the French radio manufacturers were able to show many new developments.

Valves. The principal developments have occurred in connection with smaller valves specially designed for short wave and ultra short wave working. Dealing first with small transmitting valves:

Triodes. These are made with indirectly and directly heated cathodes. Of the latter type many have oxide coated filaments, but the largest types have thoriated tungstan. Heater consumption varies between 1 and 3.3 amps at between 5 and 10 volts, and anode voltages are from 400 to 3,000, with dissipations of 10 to 150 watts. Amplification factors vary between 8 and 25, and anode impedances between 2,000 and 10,000 ohms.

Interelectrode capacities have been considerably reduced—anode-cathode capacities lie between 1.2 and 9 $\mu\mu$ F, consequently these valves will operate down to 1.04 metres. Four out of five types have their anode connections brought out to a horn, while three of these also have the control grid brought out in a similar way.

Pentodes. There are four new types, two indirectly heated and two with thoriated filaments. Heating takes 1.35 to 5 amps at 6.3 to 12 volts, anode voltages are 500 to 2000, and dissipations 120-125 watts. Mutual conductances range from 1.5 to 5 mA/V and anode impedances from 38,000 to 44,000 ohms, while interelectrode capacities are between 0.2 and 16 $\mu\mu$ F, with minimum wave lengths of 2 to 2.4 metres. Only the anode connections are brought out to horns.

Rectifiers. There are two valves specially for use in television. One is indirectly heated with 0.75A at 4V and gives 20 mA of rectified current at 7000V. The other has an oxide coated filament, taking 2.4A at 4V, and delivers 200mA for an anode voltage of 600 + 600V.

Receiving Valves. The 1940 American series includes an indirectly heated triode-hexode frequency changer, the triode portion of which has a slope of 2.8mA/V, as against 1.8 for the previous type. This greatly facilitates oscillation. There is a pentode which can be used as R.F., I.F. or A.F. amplifier by applying appropriate screen potentials. Used as first A.F. its screen potential can be controlled by the A.V.C. system.

The new output pentode has a slope of 9.5mA/V—hitherto American pentodes have had comparatively small conductances. The new valve is of special value for sets having few stages, such as 3 valve straight sets and 4 valve superhets.

For ultra short wave work there is a special valve voltmeter triode, directly heated by 0.6A at 5V, and having an anode current of 0.09mA and a slope of 0.02mA/V.

"All glass" valves for television are a recent innovation. These are indirectly heated at 6.3V. They include a diode giving 5mA at 50V with a capacity of 2.1 $\mu\mu$ F. There is a secondary emission H.F. tetrode whose anode current is 10mA for an anode and screen potential of 250V. The p.d. between heater and cathode may be as much as 150V, while the slope is 14mA/V and impedance 0.1MΩ.

An all glass pentode has four separate cathode connections, one of which is a horn at the top of the bulb.

Coils and Oscillator Units. The general tendency is in the reduction of the size of these components, owing to the limitations of portable receivers.

I.F. transformers have high permeability cores, which can be screwed in or out for trimming purposes without affecting the coupling. The adjustment is delicate because the thread is fine: it can be sealed with paint. The variation thus obtainable is 7kc/s on a nominal frequency of 472kc/s. The effective selectivity is 9kc/s for 5db. down. The tuning condensers are of silvered mica. There are also variable selectivity I.F. transformers having magnetic cores and fitted with trimmers.

Short wave oscillator units are arranged to provide band spread. These oscillators cover band widths of 1000kc/s near 19, 25, 31 and 49 metres, but the band spread gives a full scale motion for a width of 300kc/s. The increased accuracy of tuning-in effect increases the gain obtainable by some 20db. The aerial and R.F. circuits are "set" at the middle of each band, the oscillator only being spread.

Condensers. There are some small variable condensers of special interest. Some have a maximum capacity of 21 $\mu\mu$ F and a minimum of 4 $\mu\mu$ F. A set of two condensers, mounted side by side, is built up on an impregnated steatite base, with a cover of trolitul. The plates are brass and run on ball bearings and have a stop to prevent continuous rotation.

There is, this year, a new type of fixed condenser for high tension work up to 10,000V and made to carry 10A of H.F. The mica dielectric is coated on both sides with a layer of silver, which is very adherent. These sheets, each of which constitutes a complete condenser, are mounted in a water-proof steatite case and connected in series or parallel as required. These condensers are stable, have a maximum dielectric rigidity, and have dielectric losses less than 20% of those of an air dielectric condenser of equal capacity.

Electrolytic condensers have been improved as regards loss. These condensers are now finding a use in industry; a condenser of 15μF is used in the phase splitting circuit for starting single phase motors. "Non-polarized" condensers up to 100μF are also used but they must not be left in circuit for more than 2 seconds. Paper condensers must be used if they are required to remain connected across the supply.

Dials. There are some dials of unusual shape; one is in the form of a map of France, with the broadcasting stations marked roughly in the right place.

Resistors and Potentiometers. Resistors which will dissipate 2 watts are now made with a protecting varnish which keeps out water and salt air. They are made in sizes from 10 ohms to 10 megohms.

A new type of potentiometer consists of wire wound on vitreous formers, in all value from 25 ohms to 2 megohms. For larger values graphite resistances are used.

Loud Speakers. These have changed little, but the trend is towards small diaphragms and permanent magnets. Diaphragms are light, rigid and non-hydroscopic. High mu metal is used for field magnets, and high flux densities obtained. The speaker transformers are made with a layer of paper between alternate layers and give a level response from 50 to 10,000 c/s.

Pick Ups. These have been considerably improved. The suspension of the needle holder has been improved to obtain even response from 20 to 5,000 c/s. Impedance varies from 750 to 3600 ohms.

Microphones. Crystal microphones have come to the fore. They give good sensitivity with a minimum of background noise, can stand temperature changes up to 50°C, and the permanence of their response characteristic is superior to contact microphones.

Signal Generators. These are now made with push button tuning, giving 20 standard frequencies. There are also standard signal generators for U.S.W., going down to 1 metre and having a capacity attenuator giving voltages from 0.1μV to 0.1V.

The exhibition showed that French manufacturers, in spite of the needs of war, are in the forefront of progress.

Special Petrol.

An American oil company is said to have developed a new fuel with 115 Octane value. It is expected to effect 50% increase in output.

THE INVESTIGATION OF STRESSES IN A VARIABLE-PITCH AIR-SCREW.

By Dr. F. SALZMANN and A. VON DER MÜHLL. (From *Escher Wyss Nachrichten*, Vol. XII, No. 1-2.)

THE stresses set up in the hub of a modern air-screw by the great centrifugal force of over 30 tons for each blade of the air-screw (this being the approximate weight of a 4-axle railway car) were the cause of extensive research and modification in the design of the hub. As a preliminary the stresses set up had to be determined by calculation combined with experimental research, and the hub of the air-screw designed accordingly. Improvements and alterations in the design of the older hubs of air-screws were essential, particularly with the introduction of the variable-pitch air-screw, the stresses had to be reduced to a minimum, combined with the reduction in bulk of the hub. It will be seen from some examples how theoretical analysis, combined with practical research, influence the general construction and design of details. It may be stated at the outset that is not always possible to solve all problems by theoretical investigations; recourse must then be had to practical experimental research, direct measurements are then made of the stresses set up under various conditions. Again, as will be shown later, it is not always possible to measure the stresses, the problem is then solved by theoretical analysis combined with the method of interpolation. It is of interest to record that a method familiar to the physicist, the tensioned skin of soap, was applied in the laboratories of Messrs. Escher Wyss to determine the most suitable form of the hub casing. The tensions set up in the soap skin model are uniform in all directions (double surface tension). The shape of the single piece hub casing, as actually designed, was based on the soap model test, this assures the most suitable disposition of the material, and is, in fact, an ideal solution of the problem.

Any deviation from this ideal form as well as excessive uneven loading of the edges, which is inevitable in actual design, cause additional bending stresses to be set up. The stresses must be kept within certain limits, the disposition and the thickness of the material must be chosen accordingly, with due regard to the weight of the hub, which must be kept as low as possible. The first approximate values of the stresses are obtained by splitting up the hub into its elementary constituent parts, such as rings, plates, bowls, in which the stresses can be readily calculated by the elastic theory. The next step is the modification of the stress system, step by step, so as to bring the results into closer agreement with the actual case in hand. The next consideration is the investigation of the stresses set up at that point of the hub, that takes up the tensile stresses in the blades of the air-screw, occasioned by the centrifugal force. As seen in Fig. 1, the stress is transmitted through roller bearings, and is taken up by the bead (beaded flange) of the hub casing. It is proposed to describe some of the methods employed in the investigation of stresses set up in this part of the hub.

For preliminary approximate calculations, the bead and the adjoining parts of the hub casing will be considered as symmetrical rotational bodies. We shall consider the bead as a massive ring, with a cylindrical shell on one side, and a cone shaped bowl on the other. For the purpose of analysis the thickness of the walls of the cylindrical shell and of the cone shaped bowl are assumed to be constant in the plane of section. Fig. 2 represents the elementary component parts to correct scale. The three component parts are shown separately, namely, the cylinder shell, the ring, and the cone shaped bowl. The forces acting at the different sections :

$$Z,\ H_2,\ Z^*,\ H_2^*,\ N_1,\ N_1^*$$

and the moments

$$G_2,\ G_2^*,\ G_1,\ G_1^*$$

are equally indicated in the diagram. The forces and moments acting on the cylinder are referred to one cm. in length of the radius R_1, the forces and moments marked with the asterisk*, denote the forces and moments per cm. length of the radius R^* of the ring section, and lastly, Z, H_2, G_2 are the forces and moments per cm. length of the radius R_2 of the bowl section. By P^* is denoted the load taken up by the roller bearings, resulting from the centrifugal force of the air-screw blades.

Under the influence of the load the ring section is twisted through an angle δ, and the centre of the section is displaced through the

Fig. 2.

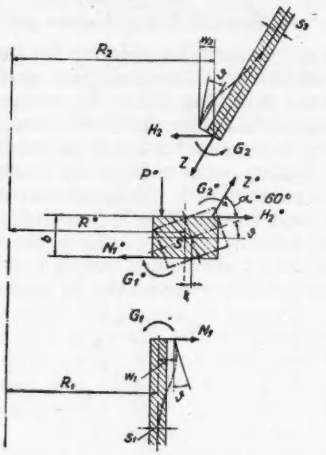
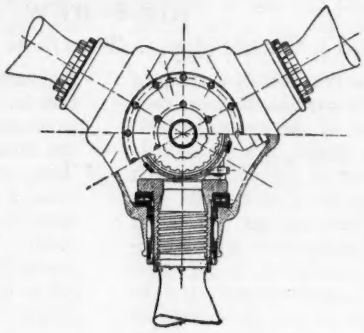


Fig. 1.



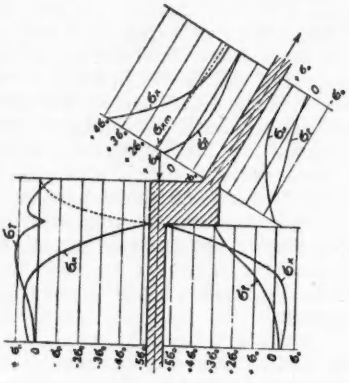


Fig. 3.

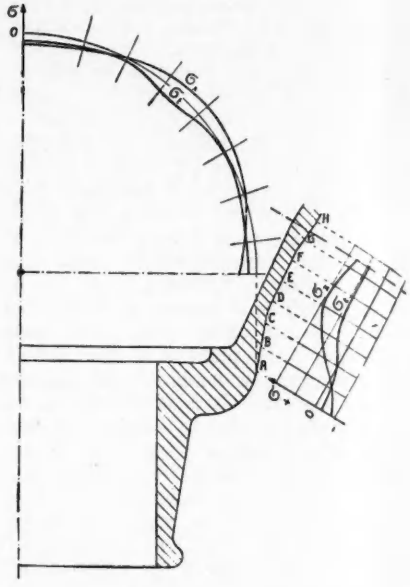


Fig. 4.

Fig. 1. Shape of the hub casing with bead. The section shows the socket into which the blade of the air-screw is screwed.

Fig. 2. Section of part of the hub casing to take socket for blades, split up into its three elementary component parts, comprising the ring (the bead) with adjoining cylindrical shell on one side and conical bowl on the other. This facilitates the preliminary determination of the stresses by calculation, the thickness of the material of cylinder and bowl being assumed uniform.

N_1, Z_1, H_2, = forces acting on the peripheral section of cylinder and bowl.
G_1, G_2, = moments at the edges of the cylinder and bowl shell.
Z^*, N_1^*, H_2^* G_1^*, G_2^*, = corresponding forces and moments acting on the ring section.
P^* = the pressure on the roller bearings, due to the centrifugal force.
—·—·— Deformation of the component parts.

Fig. 3. Course of the stresses as determined by calculation in a simplified shape of the bead of a propeller hub and its adjacent parts.

Fig. 4. Measured course of the stresses in an actual bead of a propeller hub and its adjacent parts.

Fig. 5.

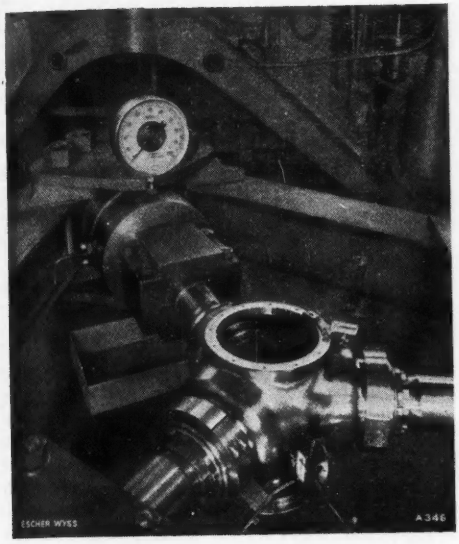


Fig. 8.

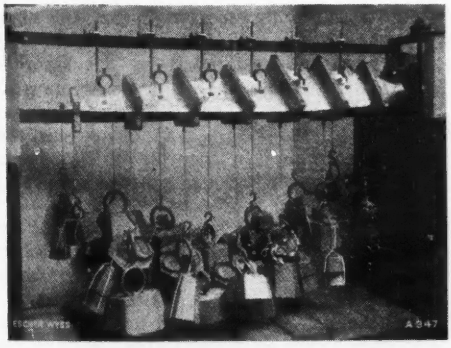


Fig. 6.

Fig. 7.

Fig. 5. The hub casing in the testing machine with the tensometers in position.

Fig. 6. Appliance for measuring the strain produced in a loading test on blades of an air-screw. The load is produced by a shearing force and a tangential force.

Fig. 7. Testing apparatus for measuring the load on the hub casing. The stresses produced by the centrifugal forces are reproduced by oil pressure.

Fig. 8. This air-screw can be adjusted when at rest on the ground. An increased range in performance is obtained, as compared with the ordinary air-screw.

distance ξ out of its original position. At the same time the edges of the cylinder and the bowl section are displaced by an amount=w_1 and w_2 respectively, the displacement being accompanied by a twist=δ.

The condition of equilibrium of the ring is given by:

$$Z^* = \frac{P^*}{\sin \alpha} \qquad (1)$$

The equations to determine the remaining statically undetermined forces are found by establishing the relation between the forces and the displacement and twist of the different sections under discussion, according to the elastic theory.

According to the theory of the cylinder shell we have the relation:

$$\frac{Es_1^3}{12(1-\nu^2)}\,\delta = x_1 G_1 - \frac{x_1^2}{2} N_1 \qquad (2)$$

$$\frac{Es_1^3}{12(1-\nu^2)}\,w_1 = -\frac{x_1^2}{2} G_1 + \frac{x_1^3}{2} N_1 \qquad (3)$$

where

$$x_1 = \frac{\sqrt{R_1 \cdot s_1}}{\sqrt[4]{3(1-\nu^2)}} = \frac{\sqrt{D_1 \cdot s_1}}{\sqrt[4]{12(1-\nu^2)}} \qquad (4)$$

and

E=the modulus of elasticity

ν=coefficient of contraction=0.3 for steel.

Other dimensional values are seen in Fig. 4.

The formulae given above are strictly true for a cylinder of infinite length, but in the present instance they are a good approximation, since the stresses diminish similarly to the fading oscillations expressed by:

$e - \frac{x}{x} \cos\left(\frac{x}{x}\right)$ or $e - \frac{x}{x} \sin\left(\frac{x}{x}\right)$ where

x=the distance from the edge.

In the case of the cone, to the deformation due to the tension Z in the direction of the cone surface, must be added the deformation occasioned by the force H_2 and the moment G_2, for which formulae can be established similar to (2) and (3).

Further equations can be established dealing with the deformation of the ring, due to the forces and moments marked with the asterisk. These latter, when multiplied by $\frac{R_1}{R^*}$ or by $\frac{R_2}{R^*}$ are equal to the corresponding values without the asterisk. Without going into details of the calculations, the results for the case under consideration are given below:

$$G_1 = 0.0729\ Z\ R_2$$
$$G_2 = 0.0375\ Z\ R_2$$
$$N_1 = 0{,}551\ Z$$
$$H_2 = 0.196\ Z \qquad (5)$$

These results hold good on condition that:

$$R_1 = 0{,}75\ R_2,\ R^* = 0{,}872\ R_2,$$
$$s_1 = s_2 = 0{,}08\ R_2,\ b = 0{,}2\ R_2 \qquad (6)$$

The results as obtained from (5) are shown in the diagram of Fig. 3 in which the falling off of the stresses towards the edges will be noticed. The mean static tensile stress at the edge of the cone, expressed by $\sigma_0 = \frac{Z}{s_2}$, is taken as unit stress in the diagram. The maximum stress occurs at the intrados of the cone shell, at a point at which the direct measurement of the stress cannot be carried out. This example illustrates the value of the theoretical analysis in determining the stresses. From the results obtained we get a good idea of the distribution of stresses, this is particularly valuable in instances like the one mentioned above, when direct measurement of the stresses at some particular point cannot be made. The peak stresses, as evaluated by this process are in reality not attained, because of the gradual transition of one elementary form into the other, and the more favourable surface configuration. The thickness of the walls, or the desposition of the material in the walls of the hub casing, depends on the fading of the bending moments, established in the theory of bowls. This also applies to non-symmetrical rotational bowls. The actual theoretical analysis of a hub casing is more complicated than in the example just given, by which it is only intended to demonstrate the methods of analysis to be applied in practice.

The illustration in Fig. 7 shows the hub casing in the specially designed testing machine. By means of three adjustable oil pressure cylinders located in the corners of the frame of the testing machine, the tension is transmitted to the hub casing through suitable pistons. The forces thus applied to the hub through the oil pressure are many times greater then these resulting from the centrifugal force of the air-screw. Fig. 5 is an enlarged view of the testing apparatus, and here also are seen the tensometers. These latter are fixed to the rings seen to the left and to the right in the

foreground, special care being taken not to impede the deformation of the hub casing resulting from the action of the forces produced by the oil pressure. By means of an ingenious contrivance some of the stresses set up on the inner side of the hub casing can also be determined.

Fig. 4 shows the distribution of the stresses as determined by measurements, and the influence of the varying thickness of the material will be noticed. It will also be noticed, however, that the distribution of stresses along the line A—H in Fig. 4, as obtained by direct measurements on the casing, in general shows good fundamental agreement with the results obtained by theoretical analysis on the elementary component parts of the hub casing. The stresses at the point A of the bead are equally shown in the diagram, it will be observed that the stress line is undulating in character, but that the amplitude of the undulations is considerable.

As a matter of course all parts of the air-screw are subjected to intensive and thorough research, theoretical analysis being coupled with practical tests on the test stand. We need only mention the statical and dynamical loading tests (Fig. 6), whirling and balancing tests. It is owing to this research carried out in a systematic way, that Messrs. Escher Wyss have been able to produce air-screws in which strength is combined with highest efficiency, as is proved by tests carried out in the wind tunnel of the Federal High School of Technology at Zurich. Last, but not least, attention should be drawn to the comparative light weight of this type of variable pitch air-screw, which is about 125 kg. for 1000 H.P. engine output, even for a big range of pitch change.

CORRUGATED PLY WOOD.

(From *Holztechnik*, No. 7, 1940).

THE corrugated ply wood is built up of plane and corrugated veneers of beech wood, glued together, as shown in Fig. 1. The veneers are by a special bending machine into sine like curves, parallel to the fibres. The corrugated veneers are further strengthened by paper being glued on them.

Tests carried out at the Research Institute for Ply Wood showed that :—

The weight was not more than 44 lbs./sq. in.

The water content was less than 12%.

The bending strength was in every direction about 1420 lbs/sq. in.

The tensile strength was in both directions about 1140 lbs./sq. in.

After having been kept wet at one side for 14 days the swelling was about $^{5}/_{64}$ in.

Thus about 43% wood is claimed to be saved, and this corrugated ply wood is particularly recommended for light constructions, insulating plates, telephone boxes, separating walls, doors, etc.

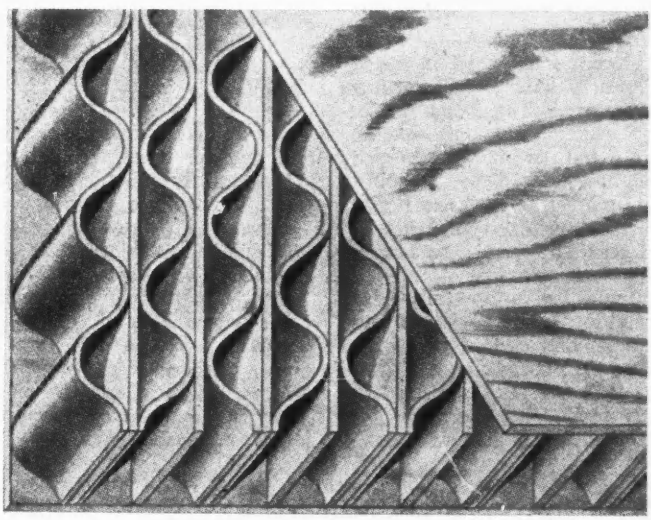


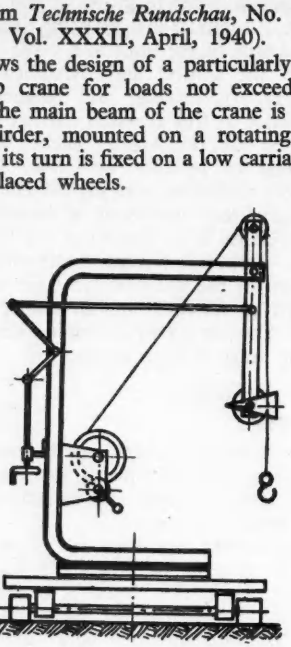
Fig. 1

PATENTS AND INVENTIONS.

A SIMPLE WORKSHOP CRANE.

(From *Technische Rundschau*, No. 16, Vol. XXXII, April, 1940).

FIG. shows the design of a particularly simple workshop crane for loads not exceeding 1½ tons. The main beam of the crane is doubly bent I-girder, mounted on a rotating plane, which in its turn is fixed on a low carriage with widely placed wheels.



CLEANSING OF BOILER TUBES WITH AMMONIA AND STEAM.

(From *Archiv für Wärmewirtschaft und Dampfkesselwesen*).

THE **Raschek** process employs steam mixed with ammonia gas to cleanse boiler tubes from sinter, crust, etc. The steam passes through a tank containing ammonia solution of 20 to 25% and is blown through nozzles into the boiler for 5 to 20 hours, the ammonia consumption being 3 to 12 cwts.

The ammonia gas is stated to neutralise as a base the different sulphuric salts and to penetrate also into the structure of the slag layer. The continous hot steam causes the ammonia to expand in the slag layer, and thus the slag layer is being broken down bit by bit.

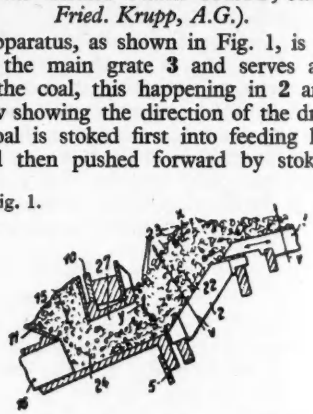
After cleansing the boiler tubes according to this process, the latter are claimed to be perfectly clean and blank. It is recommended, however, to clean the boiler first with compressed air to remove light ashes, etc., and it is stated to be an advantage to fill the boiler with cold water during the cleansing process.

MECHANICAL STOKER FOR LOCOMOTIVE BOILERS.

(From *the German Patent* 684754; *owned by Fried. Krupp, A.G.*).

THE apparatus, as shown in Fig. 1, is placed before the main grate **3** and serves also to ignite the coal, this happening in **2** and **22**, arrow **v** showing the direction of the draught. The coal is stoked first into feeding hopper **15** and then pushed forward by stoker **16**.

Fig. 1.



At the backward movement of stoker **16** the ignited top layer **23** of the coal slides backwards towards **24** and gets mixed with the coal from **15**, arrows **y** and **x** indicating the movement of the coal. Thus stoker **16** supplies a mixture of fresh and burning coal, pressed tightly towards 2.

10 is the firebox and **5** the ash chamber.

CHROMIUM PLATING OF CYLINDER LINERS.

(From *MTZ*, No. 2, 1940).

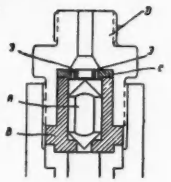
ALREADY some time ago the chromium plating of cylinder liners and piston rings has been suggested in order to reduce the wear. The results, however, were not satisfactory, especially with pistons of light alloy. It was found that the lubricating oil did not adhere sufficiently to the extremely smooth chromium plated surfaces and thus proper oil film could not be formed.

With a view to the above experience it is now suggested to apply the chromium plating in porous form, and the oil in the pores is claimed to provide sufficiently good foundation for a good oil film.

NEW FUEL INJECTION PUMP.

(From *German Patent* 687229; owned by L'Orange, Stuttgart).

FIG. 1 shows the cylinder-head of the pump block, valve **A**, which is conical at both ends, being in casing **B**. The latter is fixed to the pump cylinder by nut **D**, which holds also upper valve seat **C**. At the beginning of the injection valve **A** moves upwards, and for a short time the whole passage in **C** is free. When valve **A** reaches the upper seat **C**, it throttles the main fuel passage and this in its turn starts a low pressure wave, which may momentarily close the injection valve in the engine's cylinder-head, thus limiting the pre-injection. At the same time the pressure behind valve **A** is increased by about 350 lbs./sq. in. and the proper injection is performed through passages **J**.

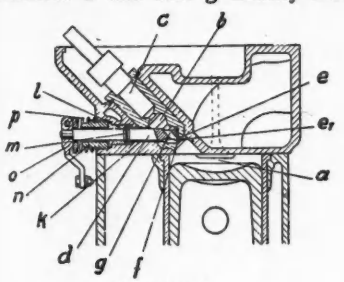


DRP 687 229 Fig. 1.

ANTI-CHAMBER GOVERNING DEVICE.

(From the *German Patent No.* 686868; owned by Daimler Benz, A.G.).

THE governing member **d** is mounted in bush **1** and its flattened end part has a bore **g**, as shown in Fig. 1. When the engine is started the injected spray passes from valve **c** through anti-chamber **b** and bore **g** directly into the main combustion space **a**. After the engine has been started, member **d** is gradually turned, until bore **g** is rectangularly situated to the fuel spray.



DRP. 686 868

Fig. 1.

GAS PRODUCER FOR SHIPS.

(From *Motor, Moscow*, No. 8, 1939).

THE Russian Scientific Institute for Wooden Rafts is said to have carried out successful experiments with gas producers for ships. The square gas producer was 1 ft. $4\frac{1}{8}$ in. long, 1 ft. $10\frac{7}{8}$ in. wide and 8 ft. $2\frac{1}{8}$ in. high, for logs of about $2 \times 2\frac{1}{2} \times 3$ in. The mean engine output is stated to be 58,6 H.P. and the starting up of the producer to require 22 min. The engine itself is started with petrol and then changed over for gas, the change being performed in about 15 secs.

COUPLING AN ENGINE WITH A GAS PRODUCER.

Condensed from *German Patent No.* 687107; owned by J. Schulz, Finsterwalde.

FIG. 1 shows diagrammatically the working cylinder of a two cycle engine. The high pressure exhaust gases are abducted through exhaust ports **12** and tube **13** into tube **14**, the rest of the exhaust gases passing through valve **15** into tube **16**. The scavanging air is mixed with steam and admitted through valve **4**. The scavanging is performed so that a part of the scavanging air-steam mixture, too, gets into tube **16**, where the same

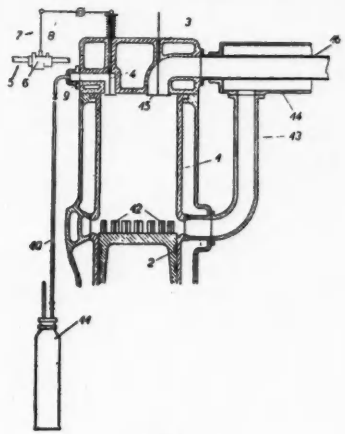


Fig. 1.

is heated by the high pressure exhaust gases. Tube **16** leads, then, further to a gas producer, which is thus supplied instead of pure air, with a mixture of air, steam and exhaust gas.

SIMPLIFIED JOINING OF WOODEN PARTS.

(From *Technische Rundschau*, No. 11, Vol. XXXII, March, 1940).

CORRUGATED steel bands, cut for suitable lengths and sharpened on one side, as shown in Fig. 1, are suggested to join timber, or other wooden parts. The bands are hammered into the wood, as illustrated in Figs. 2, 3, 4 and 5.

Fig. 1.

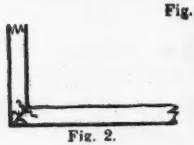


Fig. 2.

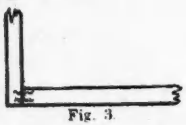


Fig. 3.

Fig. 4.

Fig. 5.

DOUBLE HEADED NAILS.

(From *Die Deutsche Illustrierte Technik*, No. 9, Vol. XLV, March, 1940).

THE double headed nails, as shown in Fig. 1, are recommended for any provisional work, in particular for the wooden frames for concrete. The nails are hammered in only till the first head, the second head sticking out sufficiently to enable, later, the easy removal of the nail.

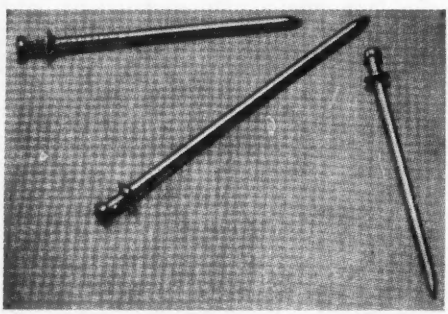
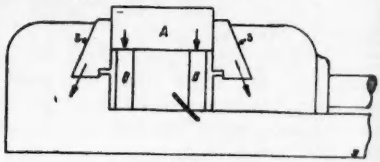


Fig. 1.

A VICE WITH SLIDING JAWS.

(From *Technische Rundschau*, No. 17, Vol XXXII, April, 1940).

FIG. 1 shows the vice in open position, and should an object be held the jaws will somewhat slide down along **b**, as shown in Fig. 2. Thus object **A** is firmly pressed down on supports **D**, as well as being sidewardly held. Fig. 3 shows the guidance of the jaws, a bolt fitting into **c** to keep the jaw in position and springs fitting into **d** to lift the jaws, which are then held by bolts fixed in **e**.



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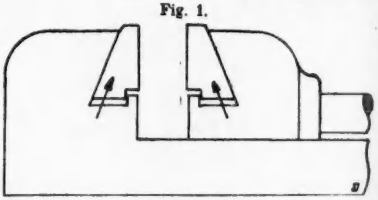


Fig. 2.

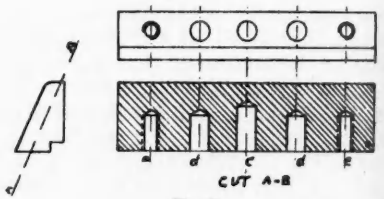
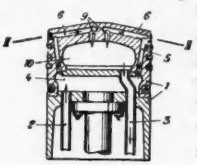


Fig. 3.

PISTON COOLING SYSTEM.

(From the *German Patent* 686359; owned by Sulzer Brothers, Ltd.).



DRP. 686 359

Fig. 1

As shown in Fig. 1, the cooling medium enters through tube **2** into space **4** and passes through bores **5** into the radial passages **6**, in the piston's upper face, wherefrom it is being led back to tube **3** through bores **9** and space **10**. Particularly good cooling effect is claimed for this system.